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## OPERATION AND TRIALS OF THE U. S. FLEET COLLIER *JUPITER*.


BY LIEUTENANT S. M. ROBINSON, U. S. NAVY, MEMBER.


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A description of the *Jupiter* will be found in Vol. XXV of the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS. At the time that description was written only dock trials had been carried out. Since then extensive trials have been held, and the methods of operation described in that article have been somewhat modified, shortening the time of reversing. For that reason the method of operation will again be described. Also a brief description of the system of interlocks on the switches will be given. The "ahead" and "astern" oil switches of each motor are mechanically interlocked so that both can not be thrown in at the same time; the oil switches and the short-circuiting levers are mechanically interlocked so that it is not possible to close the oil switches when the levers are in the "resistance out" position (the levers can, of course, be moved to this position whether the switches are open or closed); the short-circuiting levers are

locked by electric solenoids in the field of the main generator so they cannot be moved while there is field on the generator (that is, when the circuit is alive); since the installation on the ship, however, it was found to be unnecessary to lock the levers when in the "resistance out" position, so the notches were filed off, allowing them to be moved to the "resistance in" position while the circuit is alive. There are three general conditions of operating. Starting with all auxiliaries in operation and the turbine running with field on, the operation under different conditions would be as follows:

(1) Getting under way, coming to anchor, going alongside a ship, or, in general, handling the ship where much reversing might be expected.

Under these circumstances the resistances would be kept in on both motors. The operation in this case is very simple; to go ahead or astern on either motor, simply close the "ahead" or "astern" oil switch. This requires about one-half second for each motor, or is practically instantaneous.

(2) When it is desired to go standard speed:

It will be necessary to cut out or short-circuit the resistances. The ship will be going ahead. First, open the field switch of the exciter; second, move the control wheel on the dial to "slow speed"; third, move the short-circuiting levers to cut out the resistances; fourth, close the exciter field switch; fifth, bring the turbine up to the desired speed. The total time required for this operation is about thirty seconds. Most of this is taken up in waiting for the solenoids in the field of the generator to unlock and allow the short-circuiting levers to be moved. These solenoids can be adjusted to any desired interval, but as there is no reason for hurry in this operation, the interval is made long so as to insure there will be no burning of the contacts when the short-circuiting takes place.

(3) After having accomplished operation (2), a signal is received to back either one or both motors:

First, open the "ahead" switches; second, move the short-circuiting levers so as to put in the resistances; close the

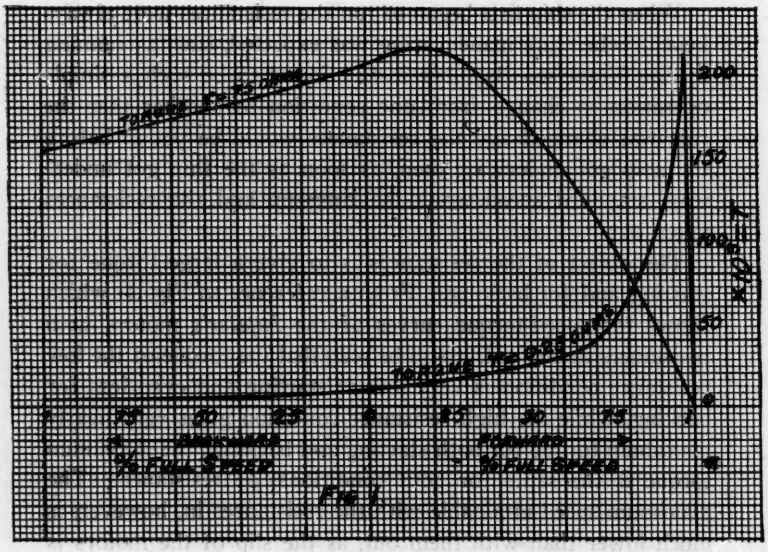
"astern" switch or switches. This operation requires less than five seconds. Originally it was intended that the field of the exciter should be opened before carrying out this operation (similarly to operation 2), but this was found to be unnecessary, as the motors became "dead" as soon as the oil switches were opened, and the short-circuiting levers could then be moved with impunity and no burning of contacts would occur. Accordingly the slots that the solenoid locks work in were filed off so that the levers could be moved from the "out" to the "in" position without waiting for the solenoids to release.

When the ship first began to run the resistances were always kept in when any maneuvering was expected, such as running in San Francisco Bay, but it was found to be such a simple operation to change from one condition to another that the rule was changed and the resistances would be put in before the ship got underway, then, as soon as the ship got underway, standard speed would be rung up and the resistances taken out.

The object of the resistances is to give a large torque for starting and for backing. From a reference to Fig. 1, which shows the speed-torque curves of the *Jupiter's* motors with resistances in and out, it will be seen that the torque of the low resistance condition is very low at starting and even less for backing, while the high resistance condition gives very nearly maximum torque for both starting and backing; the maximum torque is the same for both conditions of resistance. The maximum speed attainable with the resistances in, however, is much lower than with them out, as the slip of the motors is increased enormously. On the *Jupiter* the maximum speed with resistances in is about 9 knots; for that reason they are always taken out shortly after the ship gets underway—that is, as soon as standard speed is rung up. The efficiency of the motors is, of course, considerably reduced when operating with the resistances in.

The method of getting ready to get underway on the *Jupiter* is very similar to that of other ships. After all auxiliaries

are running, including the 35-kw. generator to be used for exciting the main generator, the steam seals are turned on the turbine glands, water service is turned on the bearings, and the oil pump is started to furnish oil for the bearings and governor relay. The control wheel at the switchboard is set at slow speed and the throttle is cracked a little and the short section of pipe to the turbine is given about a minute to warm up and then the throttle is opened till the turbine starts; the throttle is then opened slowly till it is full open and the governor has charge. The throttle is never left open with the tur-



bine stopped for the purpose of warming up the turbine, as the flow of steam through the turbine would not be uniform, and unequal expansion might take place. Warming up is really not the proper term to use, as the turbine is practically ready for load any time after starting, although it is generally allowed to run at slow speed for half an hour before getting underway and then is brought up to full speed and the emergency trip tested. This is set for 2,300 r.p.m. and is thus the

limit of the speed of the ship; this corresponds to a speed of about 16.15 knots, and is therefore ample. As soon as the turbine is started the exciter is cut in on the field of the generator and the ship is then ready to get underway.

The maneuvering qualities of the electric drive have proved to be highly satisfactory; when the resistances are in, the operation is practically instantaneous, and this makes it possible to use the engines to help out the rudder a great deal. For example, it was found to be practically impossible to swing ship for compass work in San Francisco Bay on account of interference by the shipping; the ship was stopped and the compasses were corrected and the ship then swung for compass error simply by using the engines; this would hardly have been possible with any other form of propulsion.

There are several conditions of operation that are peculiar to the electric drive. One is that, if operated at all, both motors must go at the same speed—that is to say, it is not possible to go full speed ahead on one motor and slow astern on the other; however, this is no disadvantage, or at least has not proved itself to be so on this ship; another peculiarity is that when the ship turns the inboard screw does not slow down, but keeps the same speed as the outboard; this results in a slightly larger turning circle at high speeds; it was expected that a considerable increase in the turning circle would be the result, but apparently the increased pressure on the rudder nearly makes up for the increased speed of the inboard screw. Just how the circle would be affected at slow speeds it is not possible to say accurately, as there is no data for comparison.

One of the first advantages noted in the behavior of the electric drive at sea was the total absence of racing; the governor so effectually prevents this that even with bad pitching there was no change in the speed of the motors when coming in and out of the water. Another advantage of the governor, and one that will be highly appreciated in the fleet, is the great accuracy with which any desired number of revolutions can be maintained; the control wheel once set will maintain the same speed within a small fraction of a revolution for an indefinite

period of time regardless of changes in vacuum or steam pressure.

Another advantage of this form of installation is the short time required for starting up; if standing by to get underway, the main steam pipe could be kept warmed up and all auxiliaries running slow; in this condition, five minutes would be ample time for getting ready to get underway. This would be a very valuable asset for a man-of-war.*

The *Jupiter* has had one breakdown since starting her trials. This happened in San Francisco Bay while preparing to go out for the official trials. The stationary blading in the first stage is set into an intermediate which is bolted to the turbine casing. One of these bolt heads broke off and fell into the first-stage blading, closing up both rows of moving blading and doing minor damage to the first-stage nozzles. These bolts have all been removed and the first stage of the turbine was rebladed; the bolts were replaced by fillister-head bolts which were countersunk and upset so that they cannot come out now even if they should break off. The fixed blading in the first stage does not go all the way around the turbine, and the remainder of the periphery is blanked off by a steam baffle which is also bolted to the casing. Owing to the fact that it would have been necessary to raise the rotor, these bolts were not removed; however, it is believed that no damage would result from these bolt heads if they broke off, as they are in the lower part of the casing and on the low-pressure side of the blading. The first-stage nozzles are also bolted to the steam chest. It is now recognized that it was a mistake to use bolts on any part of the interior of the turbine, but this is the first accident that has resulted from this practice, and hence it had never been changed. From the description of the turbine it will be seen that the clearances are ample all the way through, and the construction is very rugged and simple, and the turbine is believed to be very reliable. In fact, the more experience that is obtained with it, the greater confidence it imparts.

The question of the proper voltage for each speed is one

* This, however, applies to an almost equal extent to any turbine installation.

that has arisen since the ship's trials. At first the voltage used was the one that would give the best combined efficiency of motors and generator, but is was found that when turning at high speeds, with full helm, the inboard motor sometimes dropped out of step—that is to say, the torque required to drive the motor at that speed exceeded the maximum torque and the motor "broke down" and stopped—and, consequently, a slightly higher voltage is used. This really has no appreciable effect on the economy, as the voltage can be varied about 300 volts with practically no effect on the efficiency. If the voltage is reduced low enough to make a motor drop out of step the turbine is quickly slowed, the excitation of the generator increased, and the motor will come back into step and the turbine can again be speeded up.

## TRIALS.

The contract required three trials as follows:

(*a*) A progressive trial over a measured-mile course in deep water for standardizing the screws, extending from maximum speed down to a speed of eight (8) knots; about seventeen runs to be made over the course to adequately cover the range of speed desired.

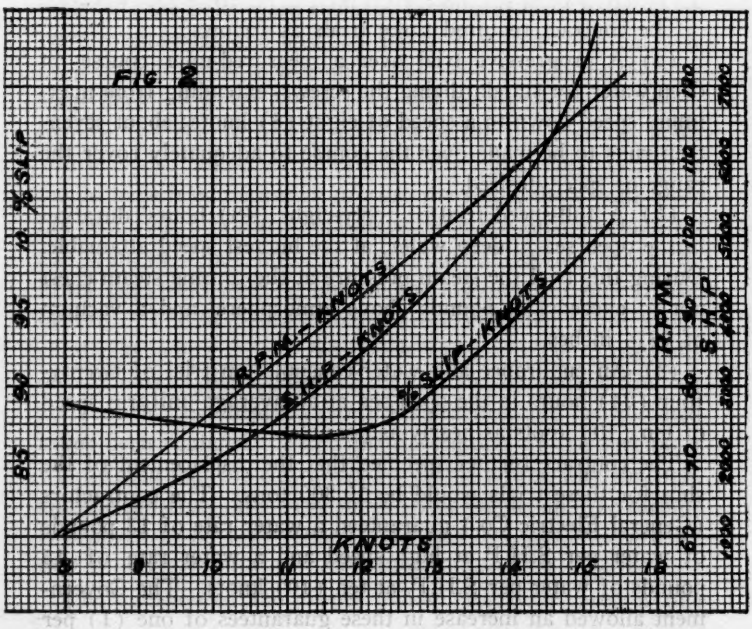
(*b*) A full-speed trial of forty-eight (48) hours' duration in the open sea in deep water, at the highest speed attainable. During this trial the steam pressure at the turbine steam chest not to exceed 190 pounds above atmosphere, and a vacuum of 28½ inches.

(*c*) An endurance and water-consumption trial of twenty-four (24) hours in the open sea at an average uniform speed of ten (10) knots as nearly as possible.

The contract further specified that the displacement of the ship for these trials would be 19,300 tons. The contractors (General Electric Co.) guaranteed a water rate of 13 pounds of water per S.H.P at high speed and a water rate of 15 pounds of water per S.H.P. at ten (10) knots. The Government allowed an increase in these guarantees of one (1) per

cent. for each ten pounds reduction of steam pressure below 190 and 5½ per cent. increase for each one inch reduction of vacuum below 28½ inches. The 48-hour high-speed trial was first run on the way to Santa Barbara. Everything worked satisfactorily during this trial and it could have been prolonged indefinitely. The average speed maintained was 14.99 knots, which was very close to the top limit of speed on account of the main steam pipe; this is so small as to be unable to take care of the steam flow for higher speeds. The water rate obtained was 11.68, which is 10.15 per cent. better than the guarantee. The coal consumption per day at this speed was 127.39 tons.

On arrival at Santa Barbara the standardization trials were carried out, seventeen runs in all being made. The runs were made in smooth water with conditions favorable. The results of these trials are shown in Fig. 2.



The 24-hour run was started immediately after standardizing and was conducted successfully. The water rate obtained was 12.32 pounds of water per S.H.P., which is 18 per cent. better than the guarantee. The coal consumption per day was 54.08 tons.

The data for the various trials is given in Table, trial data.

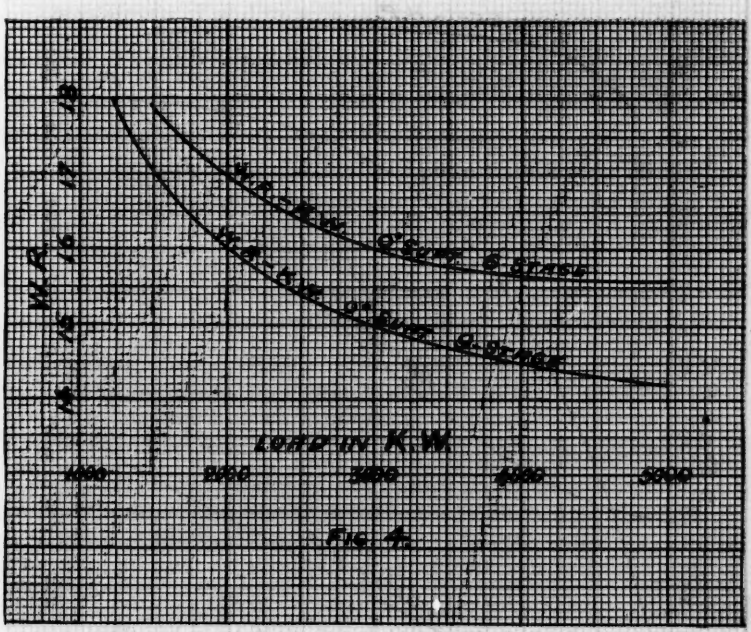


FIG. 4.

There are given in Figs. 4 and 5 some curves that were obtained at the factory in Schenectady. Fig. 4 shows the water rates of two turbines, a 6-stage turbine and a 9-stage turbine. The 6-stage machine is the one first built for the *Jupiter,* but after running tests on this turbine it was decided that improvements could be made and the new turbine was built; the results have amply justified the new construction. The 6-stage turbine has two-wheel stages throughout, while the 9-stage turbine has one-wheel stages throughout except in the first stage, which is two-wheel. The freedom from electrical

conditions which limit speed, etc., of a turbine on shore, gave the designers an opportunity to improve on previous commercial designs.

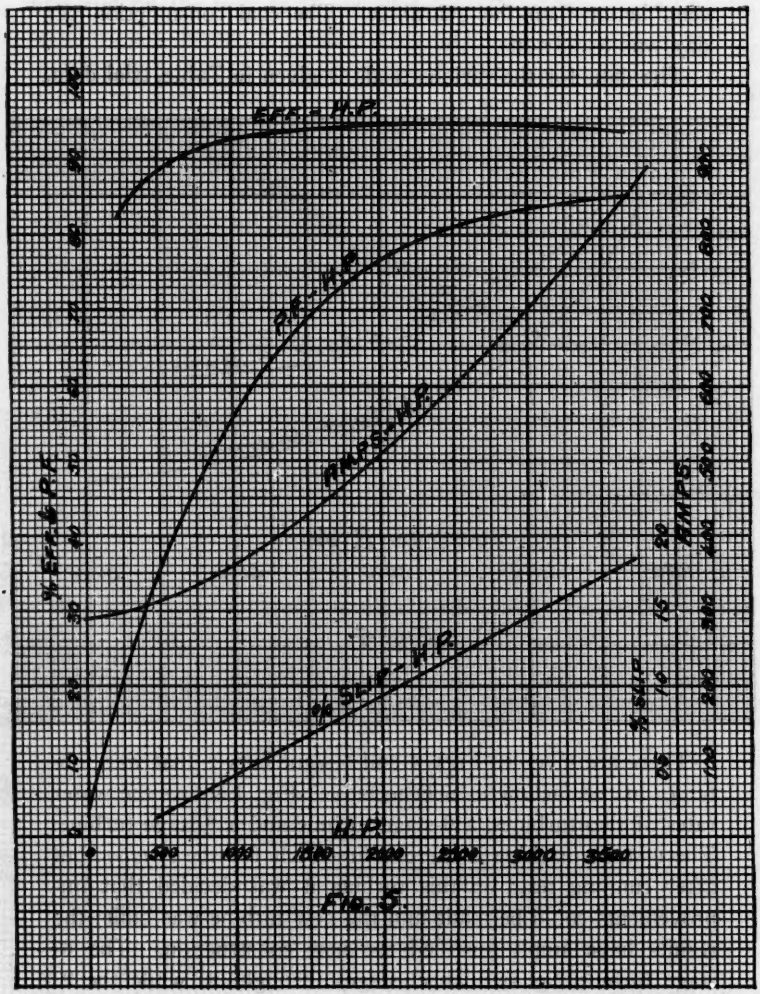


Fig. 5.

Fig. 5 shows the curves obtained in the factory for the motors. These curves are for 2,300 volts and a frequency of

33.2 cycles per second. In order to use them for other voltages and frequency use the formula,

$$x = \frac{\text{S.H.P.} \times 2{,}300^2 \times f}{\text{E}^2 \times 33.2};$$

E = actual voltage $f$ = actual frequency, and S.H.P = actual horsepower. $x$ will then be the horsepower to use in picking out data from the curves. The motors used on the *Jupiter* have wound rotors; they are used instead of the squirrel-cage type in order to be able to use slip rings and insert resistance in the rotor when backing. As in the case of the turbine, however, the new field has brought out a new motor which will be used in any further installations. This is a squirrel-cage motor with two independent windings on the rotor; the outer winding is of high resistance and the inner winding of low resistance; at high frequencies, such as are obtained in starting or reversing, the inner winding would have practically all the rotating flux by-passed around it rendering it idle; the outer winding (of high resistance) would thus give a high torque; at low frequencies, such as are obtained near synchronism, the inner winding would be doing most of the work and would make the motor efficient. This motor would make the operation of an electric installation all that could be desired in the way of simplicity and rapidity.

To sum up the case of the electric drive, it seems to be well adapted to all ships of horsepowers of 5,000 and above. Undoubtedly the ideal place for it, however, is on a battleship. The following list of advantages for it on a battleship are given:

(1) It virtually gives duplicate means of propulsion. With two turbines and two motors, one of either of them could break down and still leave a means of propulsion; this is of course true in the case of direct-connected machinery, but in the case of the electric drive, if it is a turbine that is broken down, the propulsion would still be twin-screw. The two turbines would be so designed that either would be capable of giving about 17 or 18 knots and the normal method of running would be with

one turbine and two motors. This makes it possible to overhaul the other turbine whether at sea or not.

(2) The turbines are always operated under fairly good load conditions and hence always give good economy. Since only one turbine is used at the lower speeds, the load on it will always be twice as much as for direct-connected machinery.

(3) The turbines are always operated under fairly good speed conditions, and this insures good economy of the turbines. This is due to the fact that the speed-reduction ratio is variable. This is accomplished by winding the stators of the motors for two different numbers of poles, either of which can be put into operation by simply throwing a switch. For example, if the generator has two poles and the motors are arranged to have either 30 poles of fifty poles, the reduction in the first case would be 15 to 1, and in the second case 25 to 1.

This combination of good load conditions and high speed is really the vital point so far as economy at cruising speeds on a battleship is concerned, and it is an advantage that any other form of propulsion will find it difficult to overcome.

(4) The governor absolutely eliminates all racing in a seaway, with the attendant strains on shafting and danger of throwing propeller blades.

(5) Speed can be maintained with much greater accuracy than with any form of propulsion. This is of little importance to a mechantman, but it is not necessary to emphasize the importance of this on a battleship to any one who has been in the fleet.

(6) The space taken up is less than required for other forms of propulsion.*

(7) The weight of the machinery is less than for other forms of propulsion.*

(8) The arrangement of the machinery is much more flexible than any other form of propulsion as the position of turbines and motors is not fixed.

(9) The readiness with which repairs can be effected.

* This does not necessarily hold when compared with geared turbines on the hydraulic transmitter.

There is hardly any accident that can be conceived of that could not be readily repaired by the ship's force. The rebladding of the first stage of the *Jupiter's* turbine by the ship's force shows that this part of the work can readily be handled.

(10) Due to the small size of the turbines, the upkeep would be very materially reduced.

(11) The rapidity of operation is much greater than with other forms of propulsion.

In considering the disadvantages of this method of propulsion the first thing that occurs to any one is the danger from water. However, this danger does not exist, or at least not to any great extent; the generator can be placed high enough from the floor to insure its safety as long as anything could be operated; the motors can be placed in watertight pits and made secure against water from above. All leads could be taken from the top of the generator and thus be kept well out of danger. The danger to life from high-tension current is practically *nil*, as all leads can be placed so as to be out of the way of accidental contact. The main thing to remember in considering this installation is that it is not like a power plant with leads in all directions, but is simply a reduction gear between the generator and motors. This difference is also very important when considering the damage that might arise from short-circuits; this damage would be entirely local and confined to the place where the short-circuit occurred; the generator itself would suffer no harm as it is not an enormous reservoir of power, as in the case of a power plant. The danger of such short-circuits occurring is also very remote.

In considering a change to this method of propulsion there is one question that arises at once and should be definitely settled as soon as the change is decided on. That is the question of personnel. There is no question but what the best results cannot be obtained from an installation of this kind without a personnel that understands it. The best way to get this personnel is to send all our machinist's mates to an electrical school for a few months. This would give them a ground work, and the rest they could easily pick up on the

ship. The electricians should be given a course at a machinist's mates' school, and the rating of electrician should be abolished except for radio electricians. The necessity for a mechanic in charge of the engine room would be no less than it is now, and in addition, he would have to know electricity. For this reason, the rating of electrician should be the one to be abolished. This change in the personnel would not cause any difficulty after it got started. It would be self-perpetuating. When we remember that our electricians were formerly practically all ordinary seamen who had received all their instruction on board ship, it does not seem that it should be very difficult to teach electricity to mechanics. It is very necessary that they should know it, however.

TRIAL DATA.

| | *48 Hours.* | *24 Hours.* |
|---|---|---|
| Date of trial | Feb. 14–14 | Feb. 18–14 |
| Speed, in knots | 14.99 | 10.01 |
| Draught, mean, feet and inches | 27–9 | 27–7½ |
| Displacement, in tons | 19,452 | 19,350 |
| Pressures (gage): | | |
| Main steam boilers, pounds | 193.8 | 193.9 |
| engineroom, pounds | 185.8 | 186.44 |
| turbine, pounds | 168.4 | 180.56 |
| 1st stage, pounds | 56 | 8.5 |
| Forced lubrication, pounds | 30 | 30 |
| Oil to governor relay, pounds | 80 | 75 |
| Steam seals, pounds | 5 | 5 |
| Auxiliary exhaust (absolute) | 23.19 | 23.68 |
| Air pressure to boilers, in inches of water | .712 | ..... |
| Vacuum, inches | 28.2 | 28.5 |
| Revolutions or double strokes per minute: | | |
| Starboard motor | 116.72 | 77.077 |
| Port motor | 116.72 | 77.077 |
| Main turbine | 2,130 | 1,410 |
| Wet-air pumps | 3,600 | 3,600 |
| Dry-air pumps | 112 | 113 |
| Circulating pumps | 193 | 196 |
| Feed pumps | 22 | 17 |
| Blowers | 329 | ..... |
| Temperature, in degrees Fahrenheit: | | |
| Main injection | 55 | 55 |
| Overboard discharge | 78 | 72 |
| Hottest bearing (middle) | 133 | 115 |

| | *48 Hours.* | *24 Hours.* |
|---|---|---|
| **Temperature, in degrees Fahrenheit:** | | |
| Oil from cooler | 99 | 85 |
| Feed water | 191.9 | 215.9 |
| Field ampères, main generator | 251.5 | 184 |
| volts, main generator | 101.4 | 69.44 |
| volts, main generator | 2,580.6 | 1,524 |
| Ampères, starboard motor | 725 | 395 |
| port motor | 755 | 413 |
| main generator | 1,480 | 808 |
| Kw., starboard motor | 2,692.7 | 712.5 |
| port motor | 2,833.1 | 751.3 |
| Total kw. | 5,525.8 | 1,463.8 |
| S.H.P., starboard shaft (torsion meter) | 3,603.258 | 1,053.959 |
| port shaft (torsion meter) | 3,548.677 | 961.08 |
| total (torsion meter) | 7,151.9 | 2,015.04 |
| Water consumption data: | | |
| Total pounds per hour | 105,764 | 40,066 |
| Pounds per hour auxiliaries | 22,212 | 15,246 |
| turbine | 83,552 | 24,820 |
| per S.H.P. (turbine) | 11.68 | 12.316 |
| Guaranteed water rate, pounds | 13 | 15 |
| Fuel consumption data: | | |
| Pounds of coal per hour | 11,900 | 5,050 |
| per hour per S.H.P. | 1.662 | 2.5056 |
| sq. ft. heating surface | .613 | .261 |
| Tons per day | 127.39 | 54.082 |
| Kind of coal | New River & Geo. Creek | |
| Quality of coal | Run of mine, | Run of mine |
| Slip of propellers, per cent | 9.9 | 8.74 |
| S.H.P. per square foot of grate surface | 15.9 | 4.48 |
| Knots per ton of coal | 2.82 | 4.43 |

# DESTROYER OPERATING UNDER ONE PROPELLER AND REMOVAL OF PROPELLER WITHOUT DOCKING.

COMPILED FROM REPORTS.

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A destroyer recently stripped one turbine. With the propeller in place on the shaft of the damaged turbine the vessel could not be properly handled. The propeller was therefore removed by the ship's force and vessel thereafter operated under one propeller.

As to the necessity of doing such a job is likely to appear most any time when away from a dock, the description of the removal will be of special interest.

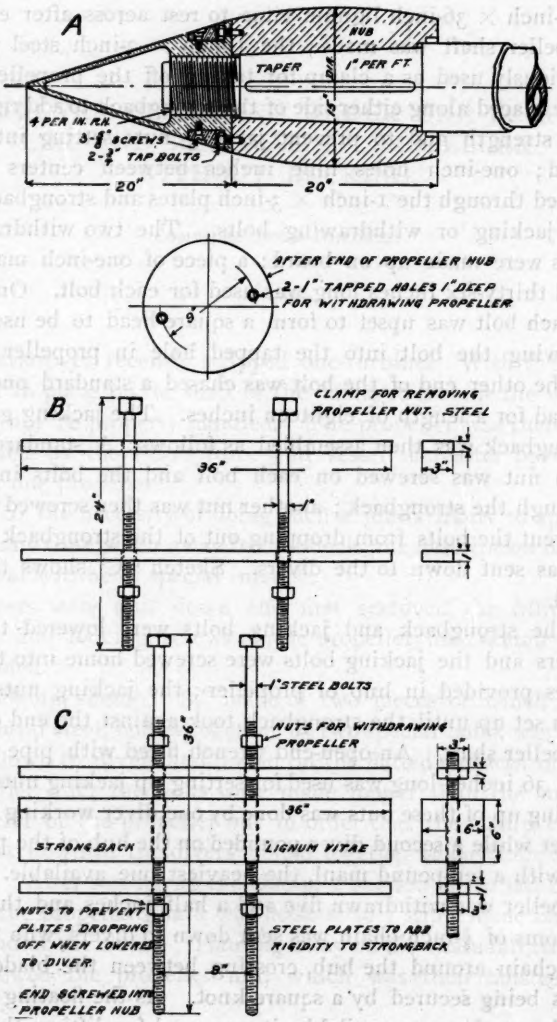
Divers were sent down and first removed the filling-in pieces and locking screws of the propeller nut, sketch "A" appended.

A clamp, sketch "B," made of two pieces of 1-inch × 3-inch mild steel, bolted together by two special bolts, was then sent down to divers and bolted on the flatted surfaces of the propeller nut; the bolts were made longer than the largest diameter of the propeller nut in order that the clamp could be sent down to the divers bolted together to facilitate their work. After setting up the clamp, a one-ton chain fall hung from the propeller guard was hooked to a strap made fast to the end of the clamp. Hauling away on the chain fall readily unscrewed the propeller nut, which was then hoisted on board.

Two one-inch tapped holes one inch deep are provided on a

nine-inch pitch circle on the after end of the propeller hub for withdrawing the propeller. A strongback made of 6-inch × 6-inch × 36-inch lignum vitae to rest across after end of propeller shaft was made; the 1-inch × 3-inch steel plates previously used as a clamp for taking off the propeller nut were placed along either side of the strongback to add rigidity and strength and to prevent jacking nuts setting into the wood; one-inch holes nine inches between centers were drilled through the 1-inch × 3-inch plates and strongback for the jacking or withdrawing bolts. The two withdrawing bolts were made up on board; a piece of one-inch machine steel thirty-six inches long was used for each bolt. One end of each bolt was upset to form a square head to be used for screwing the bolt into the tapped hole in propeller hub; on the other end of the bolt was chased a standard one-inch thread for a length of eighteen inches. The jacking gear or strongback was then assembled as follows: A standard one-inch nut was screwed on each bolt and the bolts inserted through the strongback; another nut was then screwed on, to prevent the bolts from dropping out of the strongback when it was sent down to the divers. Sketch "C" shows the rig used.

The strongback and jacking bolts were lowered to two divers and the jacking bolts were screwed home into tapped holes provided in hub of propeller; the jacking nuts were then set up until the strongback took against the end of the propeller shaft. An open-end wrench fitted with pipe extension 36 inches long was used for setting up jacking nuts; the setting up of these nuts was done by one diver working under water while a second diver pounded on the hub of the propeller with a ten-pound maul, the heaviest one available. The propeller was withdrawn five and a half inches and then six fathoms of $\frac{3}{8}$-inch chain was sent down to divers, who passed the chain around the hub, crossing between the blades, the ends being secured by a square knot. As the floating crane at the station was available, it was used for lifting the pro-


A
HUB
TAPER
1" PER FT.
4 PER IN. R.H.
3-⅞" SCREWS
2-¾" TAP BOLTS
20"
20"
AFTER END OF PROPELLER HUB
2-1" TAPPED HOLES 1" DEEP
FOR WITHDRAWING PROPELLER
9"
B
CLAMP FOR REMOVING
PROPELLER NUT. STEEL
36"
24"
1"
3"
1" STEEL BOLTS
C
36"
NUTS FOR WITHDRAWING
PROPELLER
3"
36"
6"
STRONG BACK
LIGNUM VITAE
NUTS TO PREVENT
PLATES DROPPING
OFF WHEN LOWERED
TO DIVER
STEEL PLATES TO ADD
RIGIDITY TO STRONGBACK
8"
END SCREWED INTO
PROPELLER HUB

peller; the lower block was lowered to the divers and hooked into chain around propellers and the hook moused. The crane then took in slack until the fall was just taut.

To slip the propeller off the shaft a 4-inch manila line was secured to the propeller, and the line given to a steam launch which steamed away from the ship until the propeller swung off clear of the shaft. The crane then hoisted the propeller and lowered it on deck. The divers then removed the two propeller keys from the shaft.

The work was done in a total working time of twenty-two and three-quarter hours. The work of putting on the strongback and passing the chain was probably the most difficult, and was done twice, as the first strongback used was found to be rotten, and the jacking nuts set into the wood instead of bringing a strain on the propeller; it was replaced by the strongback described.

The propeller is 80 inches in diameter and weighs approximately 1,600 pounds.

After removal of the starboard propeller the vessel ran with one propeller. Data as to increased number of revolutions and oil consumption was obtained. Unfortunately there were no torsion meters rigged, so that the horsepower under one propeller is not known; this is, however, approximated in the table by assuming that the horsepower of one engine will vary as the power-revolution curve derived from both engines.

The ship was standardized over the measured mile off Guantanamo Bay, under very favorable conditions of light to moderate breezes and smooth sea. Points on the speed-revolution curve were obtained at approximately 9.5, 12, 14, 15.5, 17.5, 20.3 knots. Three runs were made at each speed, in alternately opposite directions, the ship being carefully steered and revolutions being maintained as steadily as possible.

The results of the standardization are shown in the following table, together with the corresponding revolutions per minute under both engines, under the old speed-revolution curve:

| Knots per hour. | R.P.M. with one turbine, one propeller removed. | R.P.M. with both engines. |
|---|---|---|
| 9 | 190 | 162 |
| 10 | 210 | 179 |
| 11 | 231 | 197 |
| 12 | 253 | 214 |
| 13 | 273 | 233 |
| 14 | 295 | 251 |
| 15 | 317 | 270 |
| 16 | 340 | 287 |
| 17 | 364 | 307 |
| 18 | 388 | 325 |
| 19 | 411 | 344 |
| 20 | 435 | 363 |
| 21 | ... | 384 |
| 22 | ... | 406 |
| 23 | ... | 429 |
| 24 | ... | 454 |

In other words, at low speeds, about two knots are lost when running with one turbine at any given number of revolutions, and this loss gradually increases until at 20 knots the loss in speed is approximately 3¼ knots for corresponding revolutions per minute.

It was found that the oil consumption under above conditions compared very favorably with the oil consumption of this vessel under normal condition of steaming with both turbines. An endurance run of four and one-half hours was run with the following results:

| Revolutions per minute. | Speed. | Temperature. | | Vacuum. | Burners in use. | Pressure, burner line. | Temperature of fuel oil. | Gallons per knot. |
|---|---|---|---|---|---|---|---|---|
| | | Injection. | Overboard discharge. | | | | | |
| 250.2 | 11.9 | 78 | 104 | 27.1 | Three per boiler. Two boilers. | 170 | 150 | 28.33 |

The following comparisons at other speeds were obtained:

| | *Revs.* | *Speed.* | *Gals. oil per knot.* | *S.H.P.* |
|---|---|---|---|---|
| Both propellers, . . . | 250 | 14 | 28.6 | 900 |
| One propeller, . . . . | 250 | 11.9 | 28.33 | 450 |
| Both propellers, . . . | 214 | 12 | 26.7 | 550 |
| Both propellers, . . . | 363 | 20 | 34.8 | 2,800 |
| One propeller, . . . | 435 | 20 | 40.9 | 2,400 |

When the starboard turbine blading stripped, the turbine was locked and could not be moved; as the thrust block is forward of all shaft couplings, the starboard propeller could not be disconnected, and hence served as a drag, seriously affecting the steering. Seventeen degrees of left rudder was necessary to keep the ship on a steady course; when steaming at 15 knots or more the steering engine could not pull the rudder beyond 23 degrees, and to obtain 30 degrees the ship had to be slowed before the rudder could be put over. This resulted in a turning circle of approximately 5,000 yards diameter turning to the left. Right turns could be made, of course, without difficulty, as, in addition to the rudder, the effect of the port propeller, left-handed screw going ahead, was to assist the turn.

In order to render the ship manageable and to reduce fuel consumption, the starboard propeller was removed, as previously stated, and the following results were obtained: Four degrees left rudder is now carried at all speeds to maintain a steady course; tactical diameters were obtained at 12 knots, using 30 degrees of rudder, the diameter of left turning circle being 610 yards, and that of right turning circle being 545 yards.

Above 18 knots speed the steering engine lacks power to pull the rudder beyond 23 degrees left, while it readily pulls the rudder to 30 degrees right; this is accounted for by the port-screw current acting against the rudder; tactical diameters at 20 knots gave a turning circle of 1,300 yards turning to the left, and of 850 yards turning to the right.

## Results.

The results secured indicate that there is a material falling off in economy when only one propeller is used. The question of whether it is more economical to operate with one engine or with both at very low speeds has often been discussed. These results indicate that it is better to use both engines even at quite low speeds.

The operations also disclose the ability to go as high as 20 knots using only one engine with no serious change in maneuvering qualities. This represents about one-fourth full power.

A turbine battleship under similar conditions, without having propellers removed, was able to make about 15 knots, satisfactory ability to maneuver being maintained. This calls for about one-third full power.

The excess of power required when steaming with propellers on one side only in operation is accounted for by decrease in propeller efficiency and an increased resistance to overcome the rudder effect, the helm being put over to keep the vessel on her course.

From various cases noted it would appear that a speed approximately two-thirds of the maximum can be obtained with propellers operating only on one side. If there were no increase in resistance or decrease in propeller efficiency a speed approximating to .8 of full speed would be possible.

## ECONOMICAL PERFORMANCE OF VARIOUS RECENT NAVAL VESSELS.

The following data from trial performances showing the water and fuel consumption on different trials has been tabulated. All the vessels except *Henley* were fitted with Parsons turbines. The *Henley* has Curtis turbines.

On the destroyers it will be noted that the water evaporated per pound of oil at full power is hardly a very creditable performance. On more recent destroyers these results have been considerably bettered. The boilers and oil-burning apparatus installed are capable of giving an evaporation of 14 pounds of water per pound of oil at full power, when operated to best advantage. Attention is called to the results obtained with boilers tested at Navy Yard, Philadelphia, in this number.* The close agreement of water consumption is noteworthy, and it will be noted that it decreases as higher maximum powers are secured.

The great difference in the cruising radius as speed is reduced is strikingly shown.

Recent information indicates that the economy and steaming radius of late U. S. destroyers is markedly better than that reported on late foreign vessels tried. Exact information in regard to this matter is, however, difficult to obtain. It is doubtful whether the knots per ton at 15–16 knots will be materially improved over the best results that are now being obtained. This matter is also materially involved with the military consideration as to what efforts, weight and expenditure should be applied toward securing a large radius of action for destroyers at slow speeds.

The large radius has been obtained by the adoption of

* Oil Burning by Lieutenant Commander John J. Hyland, U. S. N.

DESTROYERS.

| ..... | Trial. | *Jenkins.* | *Jouett.* | *Henley.* | *Fanning.* | *Jarvis.* | *Beale.* | *Cassin.* | *Cummings.* |
|---|---|---|---|---|---|---|---|---|---|
| Total pounds of water per hour, all machinery. | 4 hr. full speed. | 196,383 | 195,926 | 208,379 | 193,859 | 182,224 | 185,689 | 229,127 | 240,038.25 |
| | 4 hr. at 25 kt.... | 93,088 | 93,334 | 105,698 | 106,672 | 109,615 | 104,103 | 104,693 | 109,779.75 |
| | 4 hr. at 16 kt.... | 31,180 | 30,625 | 24,416 | 29,449 | 32,231 | 34,408 | 32,059 | 33,458.75 |
| Pounds of water, all machinery, per hour per S.H.P. of main engines. | 4 hr. full speed. | 15.786 | 15.877 | 15.468 | 15.386 | 17.218 | 15.7364 | 14.963 | 14.695 |
| | 4 hr. at 25 kt.... | 15.711 | 15.955 | 16.653 | 16.040 | 18.438 | 16.6565 | 16.492 | 17.489 |
| | 4 hr. at 16 kt.... | 24.944 | 27.466 | 18.6382 | 22.059 | 24.309 | 25.5822 | 26.408 | 24.932 |
| *Pounds water evaporated per pound of oil (actual). | 4 hr. full speed. | 12.53 | 12.32 | 12.896 | 12.09 | Data unavailable. | No data. | 12.63 | 13.2208 |
| | 4 hr. at 25 kt.... | 12.36 | 12.45 | 12.40 | 11.86 | | 13.66 | 13.204 | 13.858 |
| | 4 hr. at 16 kt.... | 13.00 | 13.30 | 11.588 | 11.84 | | 13.50 | 13.055 | 14.288 |
| Pounds of oil per hour, total per S.H.P. of main engines. | 4 hr. full speed. | 1.262 | 1.286 | 1.20 | 1.273 | | No data. | 1.185 | 1.112 |
| | 4 hr. at 25 kt.... | 1.269 | 1.279 | 1.333 | 1.352 | | 1.2188 | 1.249 | 1.262 |
| | 4 hr. at 16 kt.... | 1.916 | 2.068 | 1.6083 | 1.864 | | 1.8944 | 2.023 | 1.745 |
| Square feet of heating surface per S.H.P. of main engines. | 4 hr. full speed. | 1.445 | 1.456 | 1.336 | 1.426 | 1.815 | 1.525 | 1.405 | 1.316 |
| | 4 hr. at 25 kt.... | 3.035 | 3.072 | 2.816 | 2.705 | 3.228 | 2.880 | 3.387 (4b) | 3.425 |
| | 4 hr. at 16 kt.... | 3.600 (1b) | 14.033 (1b) | 6.870 (2b) | 3.372 (1b) | 3.61 (1b) | 6.70 (2b) | 4.428 (1b) | 4.005 (1b) |
| Knots per ton of oil ..... | 4 hr. full speed. | 4.467 | 4.56 | 4.216 | 4.18 | Data unavailable. | No data. | 3.72 | 3.764 |
| | 4 hr. at 25 kt.... | 7.47 | 7.52 | 6.570 | 6.24 | | 7.35 | 6.78 | 6.78 |
| | 4 hr. at 16 kt.... | 15.00 | 15.72 | 16.976 | 14.34 | | 14.37 | 13.95 | 14.78 |
| Speed of vessel ..... | 4 hr. full speed. | 31.272 | 32.266 | 30.412 | 29.994 | 30.009 | 29.649 | 30.137 | 30.574 |
| | 4 hr. at 25 kt.... | 25.055 | 25.140 | 25.000 | 25.049 | 25.16 | 25.00 | 24.048 | 23.991 |
| | 4 hr. at 16 kt.... | 16.031 | 16.008 | 15.968 | 15.964 | 15.976 | 16.345 | 15.522 | 15.470 |
| Slip of propellers, per cent.. | 4 hr. full speed. | 23.87 | 24.70 | 24.80 | 25.11 | 21.58 | 20.56 | 23.85 | 24.495 |
| | 4 hr. at 25 kt.... | 14.30 | 14.72 | 17.17 | 17.56 | 15.97 | 14.61 | 13.31 | 13.35 |
| | 4 hr. at 16 kt.... | 9.497 | 9.72 | 12.94 | 11.97 | 10.18 | 9.47 | 8.89 | 7.935 |

*The factor of evaporation will average about 1.08.

## BATTLESHIPS WITH PARSONS TURBINES.

| ..... | Ship. | 4-hour, full speed. | 24-hour, 19 knots. | 22-hour, 12 knots. |
|---|---|---|---|---|
| Total pounds water per hour ..... | *Arkansas*... | 440,523 | 313,553.5 | 108,242 |
| | *Wyoming*... | 453,303 | 314,156 | 126,488 |
| Pounds of water per hour per H.P. of main engines. | *Arkansas*... | 15.438 | 15.227 | 24.8604 |
| | *Wyoming*... | 14.419 | 15.115 | 23.629 |
| *Pounds of water evaporated per pound of coal (actual). | *Arkansas*... | 8.6255 | 9.089 | 7.373 |
| | *Wyoming*... | 9.70 | 9.74 | 8.63 |
| Pounds of coal, total per H.P. of main engines, per hour. | *Arkansas*... | 1.79 | 1.675 | 3.372 |
| | *Wyoming*... | 1.48 | 1.553 | 2.739 |
| Pounds of coal per square foot of grate surface..... | *Arkansas*... | 35.76 | 24.159 | 15.420 |
| | *Wyoming*... | 32.61 | 22.59 | 15.01 |
| H.P. (all machinery) per square foot of grate surface. | *Arkansas*... | 19.98 | 14.420 | 4.574 |
| | *Wyoming*... | 22.01 | 14.55 | 5.623 |
| Knots per ton coal..... | *Arkansas*... | 1.083 | 1.262 | 1.83 |
| | *Wyoming*... | 1.0202 | 1.334 | 1.86 |
| Speed of vessel..... | *Arkansas*... | 21.05 | 19.4 | 11.993 |
| | *Wyoming*... | 21.22 | 19.209 | 12.266 |
| Slip of propellers, per cent ..... | *Arkansas*... | Mean 20.23 | Mean 17.31 | Mean 14.95 |
| | *Wyoming*... | Mean 17.73 | Mean 15.82 | Mean 13.12 |

*The factor of evaporation is about 1.07.

cruising engines or cruising turbines, which add from 10 to 20 tons to the machinery weights. On recent British destroyers these cruising units are not fitted, and comparatively little effort is made to secure a large steaming radius at low speeds. The extra cost of the installation is, of course, made up in a few years' service, but with the extra weight additional power could be secured.

The amount of water used for auxiliaries is not known, but on trials of the recent destroyers built at Wm. Cramp and Sons an approximation is secured by measuring all water on the *Benham* and measuring only the water used by main engines on the *Alywin* and *Parker*. The results indicate that on these vessels the following percentages of total water is used by the auxiliaries:

| | |
|---|---|
| Full speed, . . . . | 13 per cent. |
| 24.0 knots, . . . . | 13.8 per cent. |
| 15.5 knots, . . . . | 17 per cent. |
| 12.0 knots, . . . . | 26 per cent. |

From the above it will be seen that at all speeds more auxiliary exhaust is available than can be utilized in the feed-water heater, so that the power gained by utilizing the excess in the lower stages of the turbines is a very appreciable amount under all conditions of running.

The close agreement of water consumption on the battleships with that on the destroyers will also be noted. Although the total power in the case of the battleships is considerably greater, owing to the reduction in revolutions, no increase in economy per H.P. developed is secured on account of the larger unit.

There is, however, a considerably better propeller efficiency on the battleships, so that the over-all efficiency is materially better than on the destroyers. On the other hand, the weight per S.H.P. is very much less on the destroyers than on the battleships.

The returned weights indicate results as follows:

| | |
|---|---|
| Battleships: Horsepower per ton, . | . 15 to 17 |
| Destroyers: Horsepower per ton, . | . 40 to 60 |

On the battleships this weight includes all auxiliaries, ice machines and evaporating plant.

Considering the engine and boiler-room weights only, the battleship weights will run 21–23 horsepower per ton. Hence the machinery weights on destroyers are about one-half per horsepower of what they are on battleships.

# DESIGNING A FOUNDRY FOR A NAVY YARD.

By Lieut. F. M. Perkins, U. S. N.

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It is the purpose of this paper to indicate briefly the tendency of modern foundry construction and its related problems of heating, lighting and ventilation; to set forth the underlying principles governing the layout of all foundries and to illustrate in some detail the application of these general principles to the particular case of the layout of a navy-yard foundry.

The laying out of a navy-yard foundry presents a problem varying considerably from that which ordinarily confronts the designer. The chief point of difference which distinguishes it from the average commercial foundry is the great variety, both as to size and composition, of the castings it is required to produce. They vary in weight from one ounce to twenty tons, and their composition includes gray iron, steel, brass, the bronzes, the various white-metal mixtures, and all the other non-ferrous alloys used in modern engineering.

In commercial work the weight and composition of the product are the main factors in determining the design, layout and equipment of the plant. A foundry is ordinarily designed for gray iron, steel, brass or aluminum; for light, medium or heavy work, or for a combination of two or more of these requirements.

A foundry used in connection with the building and repairing of large ships and marine machinery must combine all these requirements in a degree seldom found except in a foundry for this particular class of work. The layout of such a foundry is therefore not that of a gray-iron foundry, a brass or a steel foundry, nor is it three such foundries separately de-

signed and covered by a single roof. It must be a carefully thought-out plan where unnecessary duplications are avoided and the various requirements are made to harmonize.

No question as to the relative merits of a combined foundry as compared with separate brass, iron and steel foundries has been considered. The advantages of the former are so great as to leave little room for argument. The main factors which decide in favor of the combined foundry are the lower first cost of the building, less ground space, no unnecessary duplication of equipment, lower cost of supervision, simpler transportation problem, easier routing, smaller overhead expense, and greater flexibility of organization. Specialization, which is the principal feature in favor of separate foundries, can just as easily and positively be carried out in a properly designed and well managed combination foundry.

## SELECTION OF THE SITE.

The location of the building, except within comparatively small limits, is usually determined by considerations beyond the control of the designing engineer. It is his province to accept the conditions as they are and to carefully study them, to analyze all the various elements entering into the problem and consider their effect upon the design and layout, and to determine the best shape, size and general arrangement of the building which can be made under the existing conditions and limitations.

Such a thing as a model foundry for general use, to be placed here or there, does not exist. There is no one best type. The conditions which affect the arrangement of the casting plant are numerous; they vary widely and two sets of identical conditions are seldom found. The problem is individual and one to be solved in each case by some one having a practical knowledge of foundry operation and requirements.

The foundry should be located close to the machine shops and in such a position as to make the transportation as short and direct as possible. It should also be conveniently near

the pattern shop to facilitate the handling of patterns. It should be so situated that a liberal amount of ground space will be available in the proper location for the stowage of foundry raw materials and flasks. It is essential that a broad-gage railroad track enter the foundry so that a car may be placed under the principal cranes, and it is very desirable that there be two such tracks. Other things being equal, account may be taken of the prevailing wind, and the foundry building or the cupola room and furnaces so located that the smoke and cinders will not interfere with the health and comfort of the employees of other shops or injure machinery.

The following notes on the construction and related features of foundry buildings are, considered as a whole, an outline of modern tendency and present practice in foundry construction. They were arrived at from the study of the plans and descriptions of a large number of modern foundries, constructed in the United States during the past seven years, and from the writer's experience in connection with the purchasing and laying out of the equipment for the new foundry at the Puget Sound Navy Yard.

## DESIGN OF THE BUILDING.

The design of the building must be prepared with consideration for the equipment which it is to contain. The equipment is selected to suit the work and the building must be laid out to suit the equipment. It is therefore of the greatest importance that the designer know exactly what the equipment is to be and that he be familiar with its operation. A plan carefully drawn up to meet the requirements of the equipment will decrease the operating costs and frequently lower the cost of the building.

The modern foundry is a well ventilated and well lighted fireproof structure of steel, concrete, brick and glass. It is usually a one-story building. The framework is of steel supported on concrete pillars; the walls are of the narrow-curtain type, built of steel lath and concrete or brick or a combination

of these, and the spaces between the curtain walls are of steel sash. When necessary to erect a large foundry on a limited space, a large mezzanine floor is introduced for light brass work and to provide space for some of the equipment which can conveniently be located on this floor.

If the ground space is so limited that a mezzanine floor will not provide the necessary additional floor space, a two-story or even a three-story building may be erected and laid out for convenient and economical operation. In reconstructing an old foundry building or building a new one where the old foundry building and pattern shop are adjacent, additional space can be secured by erecting the building on the site occupied by both shops and placing the pattern shop and pattern stowage on the second floor. If the foundry and pattern shop are not adjacent, a two-story building with pattern shop and pattern stowage on the second floor may be erected and the building formerly occupied by the pattern shop may be gained for other purposes, or the ground space occupied by it used for other construction.

It is sometimes possible to take advantage of the topography of the land to economize space. If the foundry is located near the side of a hill or an elevated tract of land, the railroad track and the iron yard can be laid at the height of the cupola charging door. This arrangement eliminates the use of an elevator for lifting coke and iron and effects a continuous saving in the handling of these heavy and bulky materials.

Another method of economizing space is to build a concrete basement under one bay for stowing sand and other materials. The sand is unloaded from the cars into chutes which empty into the bins in the basement. An elevator is provided for lifting the sand to the molding floor as needed.

The foundry floor should be perfectly level, dry and as hard as possible. An excellent floor in the new foundry at the Puget Sound Navy Yard has been made by laying a six-inch bed of clay, wetting it down to a slush, leveling it off and ramming and rolling it as it dried out. This provided a compact,

level sheet of clay over which was laid a three-inch bed of molding sand well rolled.

In foundries where molding space has been provided on upper floors, concrete laid over asphaltum has given good results. The main trouble experienced with molding floors other than those of dirt is in spilling molten metal on floors which have absorbed moisture. The quick formation of steam explodes the floor. Care should therefore be taken to render the floor as non-hygroscopic as possible, and this is ordinarily accomplished by laying the concrete over a layer of asphaltum or other waterproofing material. A brick floor is sometimes used, but has the disadvantage of being more expensive to repair after ruptures or explosions which sometimes occur, even on the best floors.

A number of types of roof are used on foundries with various advantages claimed for each. The principal requirements in addition to the obvious one of perfect watertightness are that it shall be durable and fireproof, and that it be so constructed as to furnish plenty of daylight and to afford the best possible means for the escape of smoke, gases and dust. The most general type in use is the monitor roof, having a large monitor of continuous sash with hinged windows capable of being operated from the floor.

Another type strongly recommended by one authority is the straight-sided roof with hooded cowls for ventilation and concrete tile for the roofing material, inset with glass for lighting.

The large monitor roof with hinged windows is recommended as being the most generally suitable, although any standard type of roof meeting the requirements previously mentioned should be satisfactory.

The proper ventilation of the foundry is an important subject. There is a constant supply of brass and zinc fumes from the brass furnaces, smoke from the fires used for skin-drying molds, dust from the floors and powdered materials used for partings, and steam and gas from the molds. In addition to the ventilation provided by the roof, a liberal percentage of the side windows should be hinged and efficient operating de-

vices installed. Hoods over furnaces and converters should be ample in size to collect the smoke and fumes in all positions of the furnace. It is a common sight in many foundries to see a furnace, when tilted to the pouring position, pour out a cloud of smoke and gas into the building.

Closely related to the subject of ventilation is that of heating. The proper heating of a foundry is just as important as that of any other shop. A molder cannot work with cold hands. This is particularly true in the molding of delicate and intricate work. A slip in drawing the pattern will often produce breaks in the mold which require as much time for repairing as for molding the entire piece.

The most satisfactory and most generally adopted method of heating foundries in commercial practice is the hot-blast system in which heated air is forced by a fan blower through suitable conduit to the various parts of the building. By this system it is possible to distribute the heat where needed and to easily reach all parts of the foundry. It is undoubtedly the best method of heating a large foundry. Special means for utilizing the heat of the waste gases from the cupolas to heat the air are sometimes used with this system, but cannot be used except where there is continuous melting. In a navy yard where a general hot-water or steam system is in operation it would be more economical to use the existing system.

If the foundry is about seventy-five feet or more in width, side heating alone will not be sufficient. The molding floor in the main bay is the most important section to be heated, and with side heating only, it will remain the coldest. Where steam or hot-water heating are to be used a nest of pipes should be carried through the building between the columns of the two inner rows. The side-heating coils should be laid close to the curtain walls. Iron or steel radiators are not suitable, as they occupy valuable floor space close to the windows and interfere with the location of molding tubs and benches.

The lighting of a foundry is equal in importance to its ventilation. It is a difficult place to keep properly lighted due to

the tendency of the walls, windows and roof to become dark and dirty. In piping up the building for air, connections should be made for air hose to be used in blowing out dust and soot from the windows.

Natural lighting should be provided for to the fullest extent. About 75 per cent. of the wall area should be glass and ample illumination through the roof provided.

The artificial lighting should consist of a general illumination of the bays with smaller lights placed where needed for the operation of the equipment. Plugs for portables should be installed on about 50 per cent. of the inner columns and where needed about the building.

Compressed air has a number of uses and is a valuable aid in the foundry. A number of connections should be provided where needed.

Fresh water is necessary for mixing and tempering molding and core sand and for numerous other purposes. Water connections should be installed liberally to avoid running out long lengths of hose or carrying water by hand. At least three sanitary drinking fountains should be provided; one near either end of the building and one near the center.

A clean, sanitary wash room is a necessary adjunct to a modern foundry. It should be equipped with metal clothes lockers, individual wash bowls, shower baths, closets and urinals. It is preferably located in a leanto centrally situated and opening directly into the foundry.

The transportation system of a foundry is one of the most important items affecting the cost of its operation. A foundry may be of the latest construction, correctly designed and carefuly laid out, but if the equipment provided for handling materials is inadequate, of the wrong type, or improperly distributed, the foundry will be laboring under an enormous disadvantage.

The transportation scheme should be studied out thoroughly in connection with the general design and location of the building, and its details decided upon before the construction work begins. The principal elements determining the trans-

portation system are the nature and size of the product; the shape, size and interior arrangement of the building; the location of the building with reference to the pattern shop and machine shop, and the location of the stowage space for foundry materials. Briefly, the requirements are that all materials used in the manufacture of the foundry products shall move in the most direct line with a minimum of handling and a maximum of allowable speed; that the various materials shall come together at the proper place and time; that undue accumulations shall be avoided; that all movements shall be progressive, or toward the final destination, and that this be accomplished without interference with the work.

The transportation system may be composed of any of the following systems or combinations of two or more:

(*a*) Cranes, bridge, wall and jib.
(*b*) Overhead trolleys.
(*c*) Industrial railway.
(*d*) Power trucks.
(*e*) Hand trucks.
(*f*) Special methods for certain classes of work.

In a foundry specializing on one class of work, special power operating devices are installed for handling materials. In a foundry making large quantities of small castings, the transportation would take the form of overhead trolleys or trucks carrying boxes. In a foundry making heavy machine tool castings, massive cranes would be necessary. In a foundry located in a city where ground space is costly and the highest production must be obtained per square foot of floor surface, special automatic conveyors for sand and castings are used to good advantage.

In a navy-yard foundry, with its great variations in size, class and volume of work, the transportation system must be a combination of means for handling all sizes of work. Special devices such as automatic sand conveyors do not belong in this type of foundry. The foundry should not be filled with equipment that will not be in almost constant use and for which there is no real or pressing need.

The main bay of the foundry should be served by two bridge cranes, one with a capacity of from 40 to 60 tons, and the other of from 20 to 25 tons, each having a 5-ton auxiliary hoist. In a very large foundry there should be three bridge cranes in the main bay, a 50 or 60-ton crane in the middle, with a 20 or 25-ton crane on either side. One of the most common causes of lost time and money in the foundry is waiting for a heavy crane. If there is much heavy work being done the large cranes are in constant demand for lifting copes, turning over and transporting molds, carrying heavy cores, and for pouring and breaking out large castings. Most large castings should be double-poured, and this operation requires the use of two cranes for a considerable time.

The side bays require one light bridge crane each, and if either side is divided by the cupola room a crane in each part may be necessary. Wall cranes on either side of the main bay are extremely useful in pouring, handling flasks, breaking out and transporting castings. In large foundries they are a necessity.

The industrial railroad system for both internal and inter-shop transportation is the most generally-used method of handling the lighter work and materials. This system is recommended as being the most flexible, simplest and generally satisfactory for handling a large variety of work. If the pattern shop, foundry and machine shop are so located that industrial railroad service between them can be installed, it will prove to be a real time saver. The flask yard should also be served by the industrial railroad, and the patterns from the pattern shop and the flasks from the flask yard should meet in the flask repair room. Here the bars and chucks of the cope can be changed to fit the pattern and the two sent on their way together. This saves the flaskmaker's time, permits this work to be done where the tools are provided for doing it instead of on the foundry floor, and the molder receives the flask and pattern ready for molding.

The iron yard, stowage for non-ferrous scrap, and bins for the various bulky foundry materials such as coke, molding

sand, core sand, fire clay, limestone, etc., should be laid out with careful consideration for their transportation to the points where they are to be used. Iron, coke and sand should be unloaded from the cars in which they are received directly into their stowage spaces. The space laid out for material stowage should be served by a crane for unloading and handling materials, and an industrial track or monorail for carrying them to the places where they are to be used. The crane should be equipped with a clam-shell for handling sand and a lifting magnet for handling iron.

The internal transportation must provide for efficient service from the core room, cupolas, converters and furnaces to the molding floors; from one of the cupolas to the steel converters, and from the molding floors to the cleaning room. To facilitate this internal transportation the molding space should be laid off in individual molding floors, each floor being numbered to assist in routing the work. Properly laid-out gangways for receiving castings should run along these floors. Each floor should have a small rack at one end designed for holding a proper supply of clamps, wedges, gagers, nails, rods, molders' tools and parting sand. These items are all in constant use by every molder, and their convenient and uniform location will save a great deal of the molder's time. Three or four steel racks for the larger clamps, slings, hooks, turnbuckles, feeding rods, skimmers, core straps, beams and other articles of less frequent and general use should be so disposed as to properly serve the molders. A small room should be provided for heavy equipment which is used only occasionally. In this class are included spindles, steps, sweeps, arbors, etc.

With such a layout providing a definite working space for the molder, his tools and equipment readily accessible, old flasks readily removed, and definite stowage space provided for all equipment, the foundry is in a condition to operate economically and to produce a maximum output per square foot of molding area. The cost of supervision is reduced, the routing of the work made easier, collecting and transporting

of castings simplified, and the floors more readily prepared for molding after the pouring and shaking out.

Order and neatness, so long unknown to the foundry, have become a necessity if the work is to be produced economically. A neat, orderly, convenient shop not only facilitates increased production, but also has a moral effect upon the workmen which raises the tone of the shop and bears direct results in the quality and quantity of the work produced.

In Figure 1 is shown the plan of a large, gray-iron, converter steel and brass foundry. Its overall dimensions are 135 feet × 360 feet, and its approximate daily capacity is 50 tons of iron, 6 tons of steel and 6 to 10 tons of brass. Under pressure this capacity could be appreciably increased.

It is divided into three bays; the main bay 65 feet in width, and the two side bays 35 feet in width. Sixty-five feet is a convenient width for the main bay. It permits two parallel rows of molding floors to be laid out with room between the two rows for a gangway and industrial track.

The cupola room divides one side bay into two parts, in one of which is located the steel foundry and in the other the core ovens, core room and flask-repair room.

In about the center of the opposite side bay are located the brass furnaces, crucibles, scrap bins, scales, etc., for the brass foundry which is on one side of these furnaces. The other side is reserved for light iron molding on bench and machine. It will be noticed that this arrangement effectually separates the different classes of work.

The foundry yard for the stowage of pig iron, scrap, molding sand, coke, etc., is adjacent to the cupola side of the foundry and extends nearly the entire length of the building.

This space is served by a standard-gage spur track for the delivery of materials, a 10-ton crane with a 50-foot span for unloading and handling and a 24-inch industrial track for delivering to various points within the foundry. Heavy flasks would also be stowed under the crane. This space may or may not be covered. A light-weight, well-lighted roof is recommended.

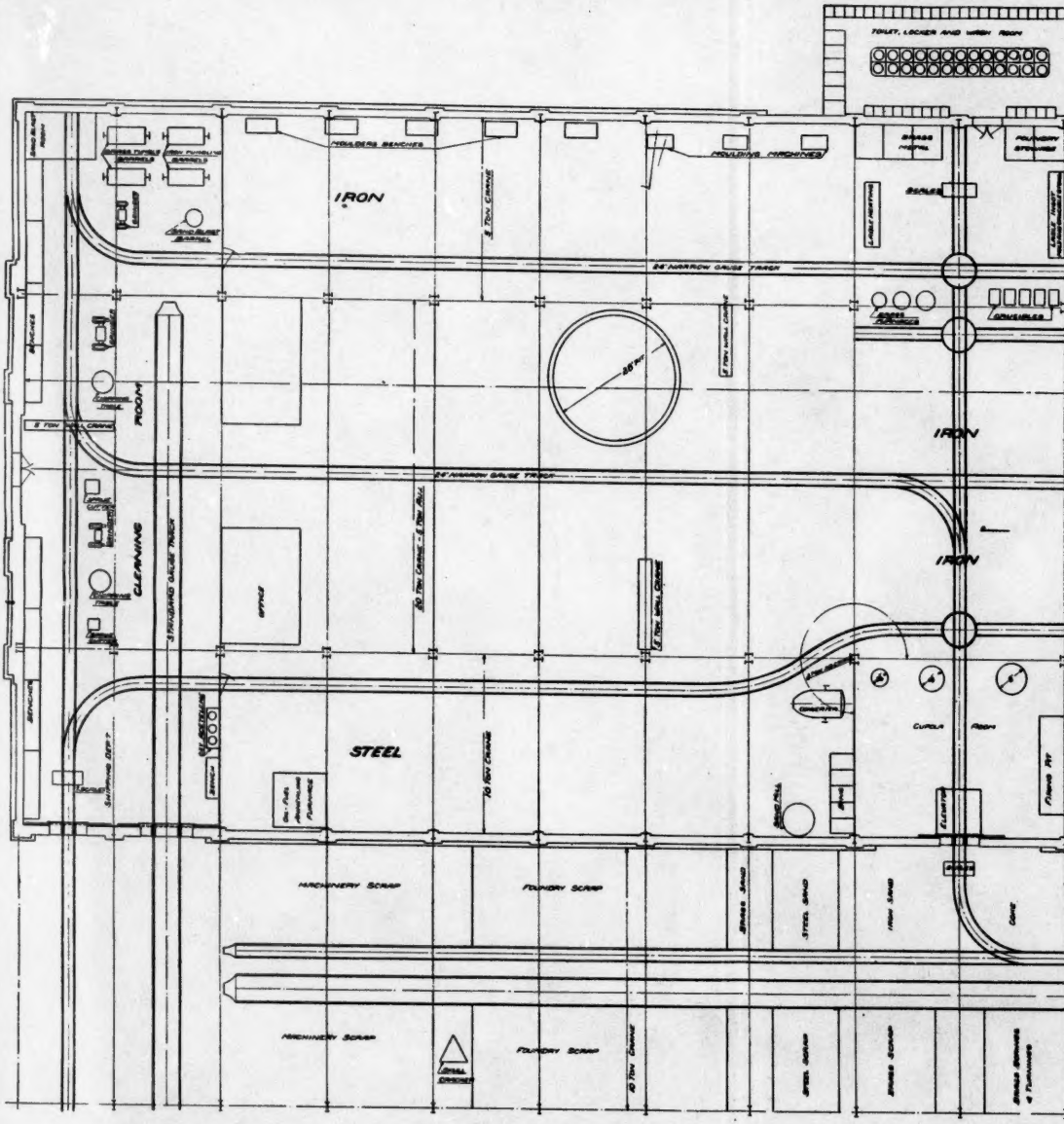

TOILET, LOCKER AND WASH ROOM
MOULDERS BENCHES
MOULDING MACHINES
IRON
5 TON CRANE
24" NARROW GAUGE TRACK
IRON
IRON
CLEANING
ROOM
STEEL
10 TON CRANE
CUPOLA
ROOM
MACHINERY SCRAP
FOUNDRY SCRAP
BRASS SAND
STEEL SAND
IRON SAND
CORE
MACHINERY SCRAP
FOUNDRY SCRAP
10 TON CRANE
STEEL CINDER
STEEL CINDER
BRASS SCRAP
BRASS SCRAPS & TURNINGS

FIG. 1.

BRASS
CORE ROOM
CORE BENCH
FLASH REPAIR ROOM
STANDARD GAUGE TRACK

G. I.

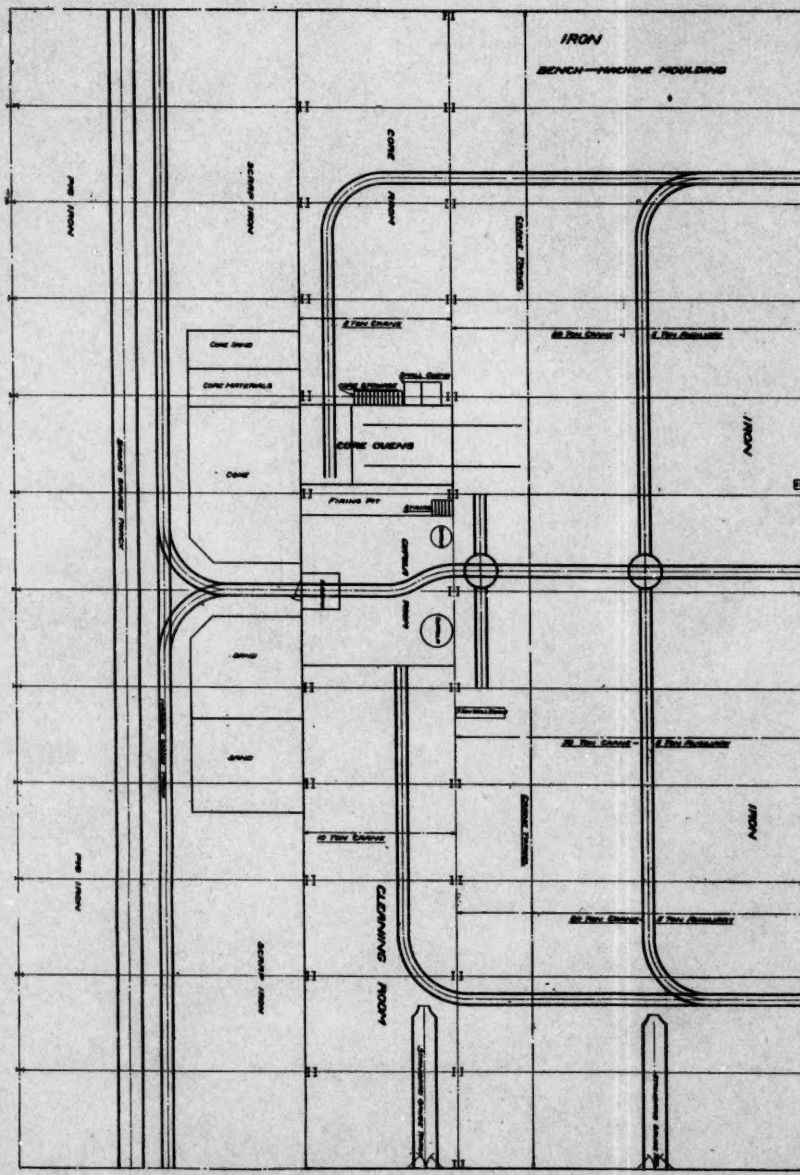
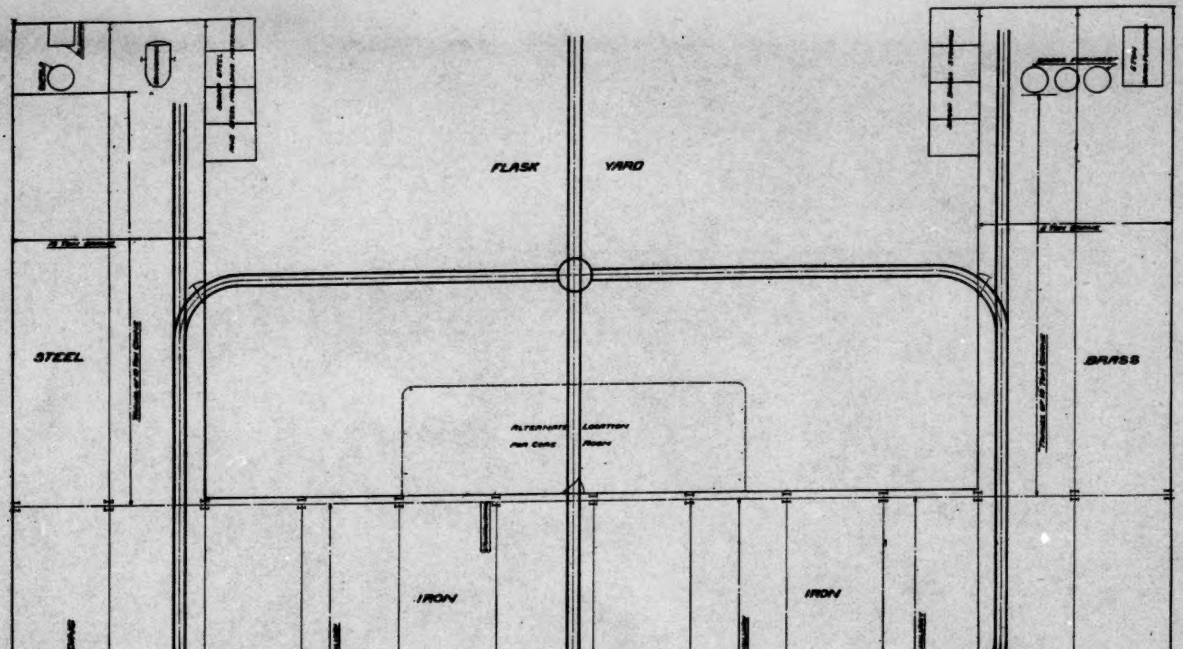



FIG. 2.




2.

The cleaning room, which is 40 feet in width, extends across one end of the building. Industrial tracks from the various molding floors lead directly into the cleaning room. The interior wall of this room extends only to the height of the crane columns and is left entirely open in the middle for a space of 20 feet. This permits one of the 20-ton cranes to enter the cleaning room and place heavy castings on a flat car, where they are cleaned without further handling. The outer wall of the cleaning room carries a 5-ton traveling wall crane.

The industrial tracks from the pattern shop and flask stowage meet in the flask-repair room, where the cope bars are adjusted to the patterns. Flasks being returned to the yard also pass through this room, where any necessary repairs are made.

Figure 2 represents the plan of a foundry arranged for location on a square, or nearly square, tract of land. In this case the overall dimensions of the building are 210 feet × 230 feet.

There are four natural divisions: the front bay, main bay and two side bays. The front bay, which is 30 feet in width, contains the core room, cupola room and cleaning room. The main bay, 80 feet in width, is used entirely for iron molding. The side bays are each 40 feet × 100 feet; one being used for brass and the other for steel molding. The space enclosed by the main bay and side bays provides stowage space for flasks and scrap metals and materials for the steel and brass foundries.

A cupola, elevator and small charging floor are installed alongside the steel converter. A converter recently placed on the market melts the iron as well as blowing it down. This type of converter, if it proves successful, would do away with the additional cupola and elevator.

The crane equipment consists of the following:

1 50-ton crane in main bay.
2 20-ton cranes in main bay.
2 5-ton wall cranes in main bay.
1 10-ton crane in steel foundry.

1 5-ton crane in brass foundry.
1 10-ton crane in cleaning room.
1 2-ton crane in core room.

In addition, there is a 10-ton crane having a 60-foot span which serves the iron yard. Cars are run into this yard on a standard-gage track, and iron, coke and sand discharged by the crane.

An industrial railway system, as shown, provides efficient transportation for delivering iron and coke to the charging floor, sand to the molding floors, castings from the molding floors to the cleaning room and flasks from the flask yard to the various floors. Large cores may also be delivered by the industrial cars; the track in the core room terminates in one of the core ovens.

A standard-gage track enters the cleaning room and also the main bay.

The general arrangement of this foundry provides for easy separation of the various classes of molding and affords a good solution of the transportation problem. Additional floor space for molding may be secured by locating the core room opposite the cupolas in the space enclosed in the dotted lines and using the space in the front bay for bench and machine work in iron.

Figures 3, 4 and 5 represent, respectively, the ground plan, mezzanine-floor plan and cross section of a foundry designed for a large capacity but covering a comparatively small ground area. A modern one-story foundry building designed to meet the needs of a navy yard, together with the stowage for scrap metals, pig iron, coke and flasks will occupy an area of from 50,000 to 75,000 square feet of ground. The foundry herewith described occupies a total ground area of 30,000 square feet, including the stowage space for flasks, iron, coke, scrap metals and all foundry materials. Its capacity is 30 tons of iron, 6 to 10 tons of brass, and 6 tons of steel per day.

From figures 3 and 5 it will be seen that the building is divided into three bays. The main bay is 65 feet in width and is the molding space for iron floor and pit molding. The

smaller of the two side bays is 30 feet in width, and is used for bench and machine molding in iron. The other side bay, which is 40 feet in width, contains the cupola room, which divides this bay in two parts. One part contains the mold-drying and core ovens and core room, and the other contains a 2-ton converter and the steel molding floor.

The cleaning room, 25 feet × 95 feet, is located at the end of the iron floors.

The cupola-charging floor is 25 feet from the lower floor level and is continued over the entire length of this bay and across the ends of the building as a mezzanine floor. That part which is over the cupola room and core room is designed for a load of 2,000 pounds per square foot and provides stowage space for pig and scrap iron.

The brass foundry occupies the mezzanine floor over the cleaning room and steel floor. An opening is left in the mezzanine floor for the hood and blast from the steel converter. The mezzanine floor at the opposite end of the building provides stowage space for flasks.

Between the charging floor and cupola-room floor and at a height of 14 feet above the latter, is a floor which is walled in and affords a location for the blowers. Here they are conveniently located away from all lines of traffic and in such a position as to permit of short leads of air conduit with easy bends.

Immediately adjacent to the 40-foot side bay is an elevated railroad laid on heavy steel trusses supported on one end by a concrete wall and on the other by the building columns, specially reinforced for the additional load. This track runs the entire length of the building. Coke, pig iron and scrap iron are discharged directly from the cars in which they are received through a number of doors to the coke bin and iron-stowage floor. The trusses supporting the track are covered with steel lath and concrete, and the space immediately underneath the track is divided into a number of concrete bins for molding sand and other foundry materials. Trap doors in

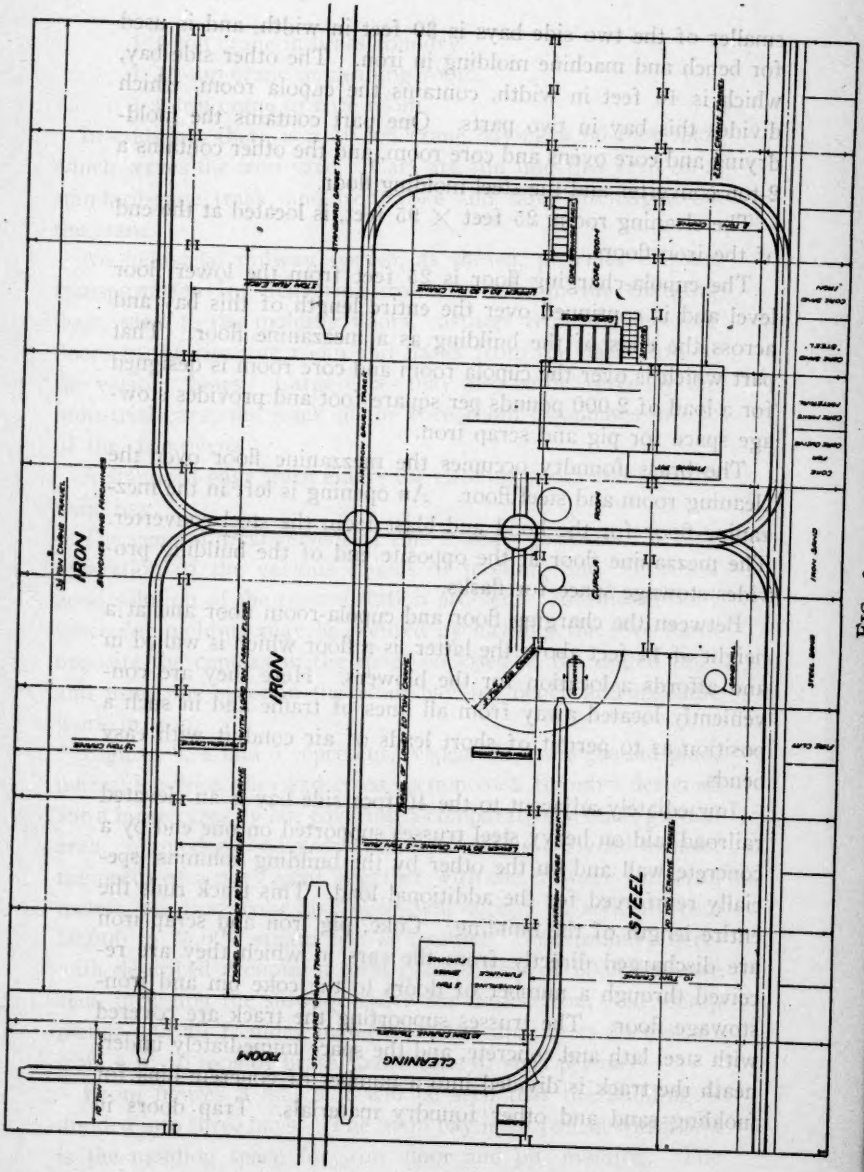

IRON
IRON
STEEL
CLEANING ROOM
CUPOLA
CORE ROOM
STANDARD GAUGE TRACK
NARROW GAUGE TRACK

FIG. 3.

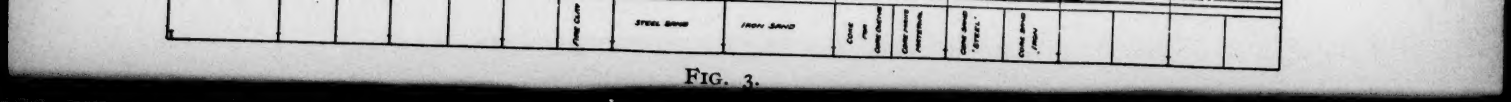


FIG. 3.

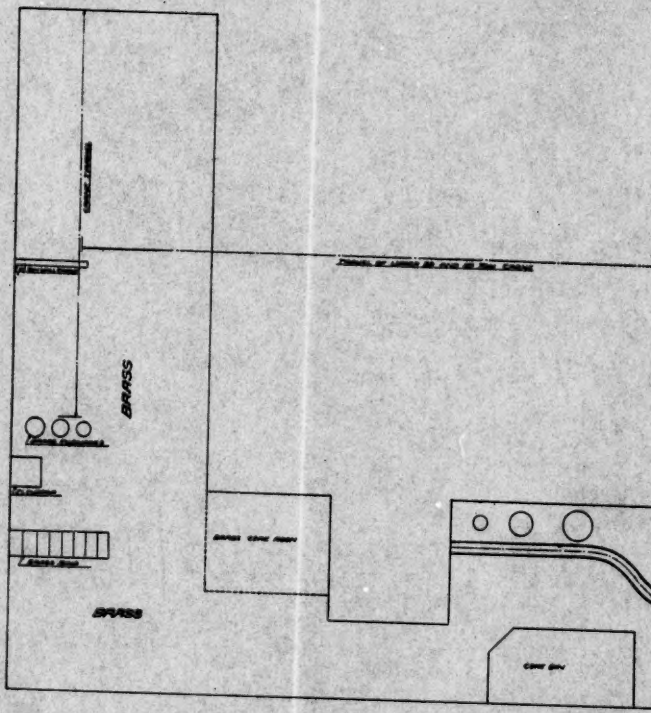



FIGURE 4-MEZZANINE FL

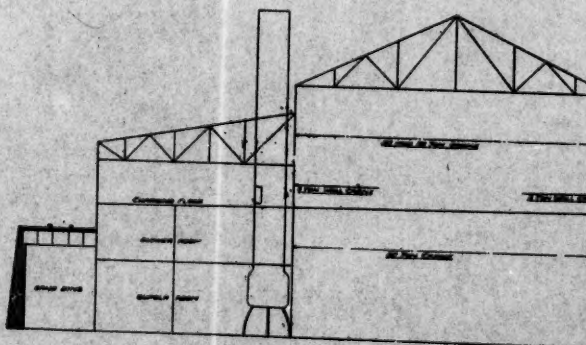


FIGURE 5
CROSS SECTION THROUGH CUPOLA ROOM

-MEZZANINE FLOOR



CUPOLA ROOM

the roof permit these materials to be discharged directly from the cars into the bins.

There is no wall between these bins and the adjacent side bay. A wall here is unnecessary and its omission greatly facilitates transportation.

It will be noted from a careful study of the plans that this layout lends itself to quick and easy transportation. The transportation equipment consists of the following:

1 50-ton crane, 5-ton auxiliary in main bay.
2 20-ton cranes, 5-ton auxiliary in main bay.
2 5-ton cranes, wall in main bay.
1 3½-ton crane in side bay.
1 10-ton crane in steel foundry.
1 2-ton crane in core room.
1 10-ton crane in cleaning room.
1 2½-ton wall crane, in brass foundry.
1 5-ton jib crane for transferring iron from cupola to converter.
1 elevator from cleaning room to brass-foundry floor.
Industrial track as indicated on the plans.

Coke, pig iron, scrap iron, molding sand, core sand, fire clay, limestone, core binders, and all the other various foundry materials may be discharged from the cars in which they are received in the yard, directly into stowage spaces within a few feet of where they are to be used. The cleaning room is so located that castings from the iron and steel floors pass in a direct line through the cleaning room on their way out of the foundry. Brass castings are lowered in the elevator directly into the cleaning room. Gates, sprues and returns for iron, steel and brass, which accumulate in the cleaning room, are placed in flat-wheeled buggies and lifted by the elevator to the mezzanine floor and returned to their stowage spaces. Flasks may be readily obtained and quickly delivered to the molding floors. Heavy castings may be loaded onto a flat-car, run into the cleaning room, cleaned while on the car and shipped out. Brass castings weighing several thousand pounds would be cast on the lower floor. Provision is made for quickly

transferring the metal from the brass furnaces to the lower floor by means of the 2½-ton wall crane and upper 20-ton crane, or, if desired, the large Mercs furnace on the lower floor may be used.

The roof of the mold-drying oven and core oven should be used for stowing slings, hooks, spindles and other equipment, which finds only occasional use.

The foregoing brief descriptions with their plans represent three types of foundry layout. The overall dimensions are, of course, subject to alteration for the purpose of increasing or decreasing the capacity of the foundry. These plans are not presented as being the most suitable for any given navy yard, but rather as examples fulfilling the principal requirements of all navy-yard foundries. It is probable, however, that one of these general types, modified in its details to meet local conditions, could be adapted to suit the particular requirements at any given yard.

# THE BREGUET EJECTAIR.

By M. M. Delaporte, Manager of the Paris Breguet Workshops.

Translated by A. B. Lep. Mesnev, Associate, from "Bulletin de L'Association Technique Maritime.

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Note.—This proposition is of somewhat similar character to the "Kinetic tank." It appears quite possible to apply this to marine work, and in any case it discloses an interesting experiment.—H. C. D.

Dry rotary air pumps, of which various designs in recent years have been made with a view of their application to the condensation of steam, have made possible the realization of very high vacua and also have made it possible to secure the maximum vacuum corresponding to the temperature of the injection or discharge water.

We should recognize particularly that in its application to marine work these pumps do not permit of simple installation.

The idea struck me, then, that equivalent results could be obtained without the use of moving mechanisms, but by the aid of steam ejectors only. Certain calculations and thought have encouraged me to take this view again, although others had already been engaged in it without success.

The placing of ejectors in series, which is indispensable in consequence of the great ratio of compression sought, is an old idea, which M. Maurice Leblanc has already brought up in the course of his remarkable and productive studies on condensation. The age of this study alone explains the failure of the early experimenters. At the time of their trials the dry-air pump had not made its appearance and the work of condensation was done by one air pump only. The difficulty

resulting from this modus operandi has been set aside for the up-flow ejector.

The problem of the down-flow ejector is not solved by simply considering the production of vacuum in frigorific machines, machines from which it is sufficient merely to extract a gaseous mixture already very deficient in air.

It was necessary to condense the steam issuing from the first or up-flow ejector, as otherwise too much load would have been placed on the down-flow or secondary ejector. Furthermore, air and water had to be extracted simultaneously from the auxiliary condenser placed between the two ejectors, for the purpose of condensing the steam from the first ejector. Though the quantity of water was relatively small the down-flow ejector was not effective in accomplishing its work satisfactorily. The operation was costly, and the poor results obtained with the ejector in series was proven.

The subsequent researches were made with a combination of steam ejectors and stand pipes. The introduction of the pipe was an alternative proposed with a view to improving the whole system. High vacua were obtained, but the amount of air extracted from the apparatus was small in comparison to the power absorbed by the pipe. As a matter of fact, in using ejectors in series with a water stand pipe it would appear that the latter is really detrimental, for the same reason that on very large systems of drainage any choking in the system of flow will reduce the delivery and require a greater amount of power to obtain the same result.

Up to now, when we were satisfied to impart a moderate flow of water in the water pipe by means of atmospheric pressure only, the combination of ejectors and water pipes permitted the application of an air extractor of such small size as to limit the use to frigorific machines only. When desirous of utilizing such a combination for extracting air from condensers, it has been necessary to have recourse to larger volumes of water and higher speeds and consequent greater expenditure of energy. The problem for the extraction of air by means of steam ejectors is really simple. The

ejector is a marvellous apparatus for the compression of gaseous mixtures, but its duties should be limited to that and should not be required to perform other work. The rest of the problem is to provide some process of extracting water from the auxiliary condenser.

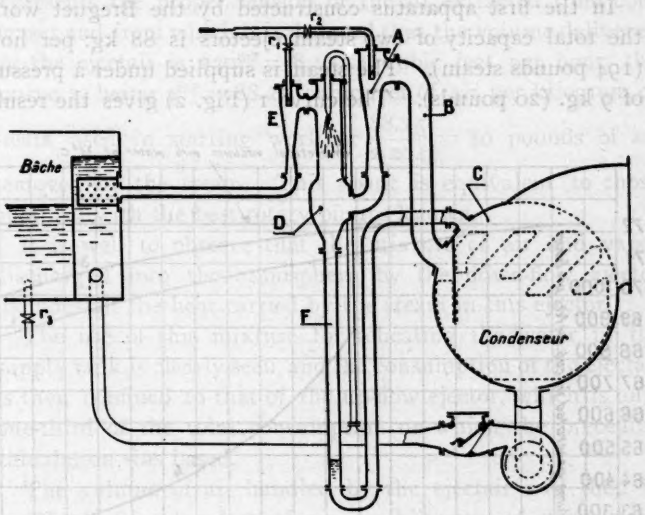



FIG. 1.

Fig. 1 represents the arrangement which I have selected. A first ejector A, or up-flow ejector, draws through a pipe B from the condenser C, and discharges the mixture of vapor and air ejected into a small auxiliary condenser D. The air is discharged from the latter condenser by a second steam ejector E, or down-flow ejector, while the water, which has a tendency to accumulate in the auxiliary condenser D, is directed through a U-shaped pipe F, towards the circulating pump of the main condenser, either directly or in its passage through condenser C. The shape of pipe F answers a triple purpose, viz: to permit the difference in pressure in both condensers to remain constant, to insure free circulation of water and to prevent the air escaping. We obtain, then, a

condensing system in which the only pump required is a reciprocating or centrifugal pump and which, being very simple and strong, gives good results.

The two steam ejectors and the auxiliary condenser have been called the "Ejectair."

In the first apparatus constructed by the Breguet works the total capacity of two steam ejectors is 88 kg. per hour (194 pounds steam). The steam is supplied under a pressure of 9 kg. (20 pounds). The curve 1 (Fig. 2) gives the results

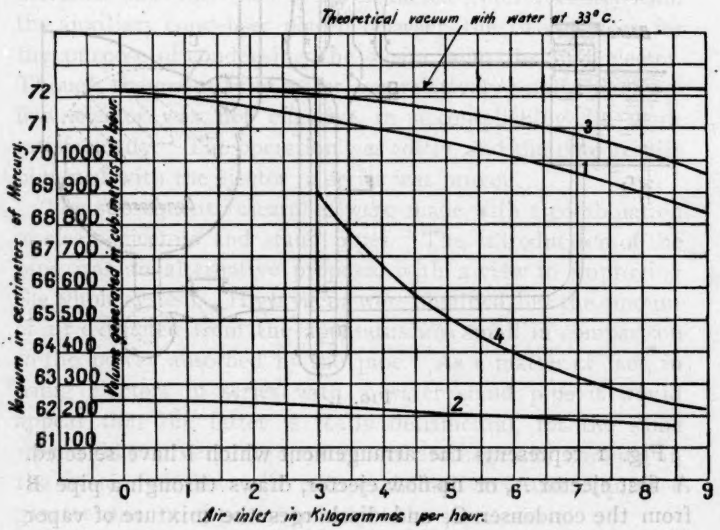



FIG. 2.

1. Vacuum curve with water discharging from condenser at 33° c.
2. Corresponding volumes generated.
3. Vacuum curve, water discharging from condenser at 33° c. and the circulating water at 18° c.
4. Corresponding apparent volumes generated.

obtained with this ejectair acting on a condenser with a mixture of $100^{m3}$ (3,531.5 cubic feet) of water per hour, in which the air supply of varying quantity was effected by means of jet pipes of suitable size. One jet pipe $3^{mm}$ diameter (0.118 inch) gave an air supply of 6 kg. per hour (13.25 pounds). The form of the curve 1 gives a supplementary air supply of

0.5 kg. per hour (1.10 pounds), due to the injector water and from the various joints and stuffing boxes in the installation.

It may be seen that for a total air supply of 2.5 kg. (5.5 pounds) per hour, the difference of the actual from the theoretical vacuum is $7^{mm}$ (.275 inch) of mercury. This value represents the proper tension of the air extracted from condenser and from which it is deduced that the volume delivered by the ejectair is $230^{m3} = 8,122.4$ cubic feet per hour, the (curve 2) being $2^{m3} = 88.25$ cubic feet of air per kilogram of steam used in starting work, or $\frac{88.25}{2.204} = 40$ pounds of air removed by the steam. This result is equivalent to those obtained with the best rotary pumps known.

It is well to observe that the mixture of air and vapor discharged into the atmosphere by the down-flow ejector includes all the heat carried by the steam in this ejector.

The use of this mixture for reheating the water in the supply tank is clearly seen, and the consumption of the ejectair is then confined to that of the up-flow ejector, which is only one-third of the total consumption on which the preceding calculation was based.

The volume of air handled by the ejectair will then be $7.5^{m3}$ (264.82 cubic feet) of air per kilogram of steam used, or $\frac{264.82}{2,202} = 120$ pounds of air removed per 1 pound of steam. Better results may be obtained by using more powerful ejectairs when contemplating the placing of 3 ejectors in series.

As shown in Fig. 1, It may be necessary to contemplate the possibility of discharging into the condenser a certain amount of water from the supply tank, by means of the valve $r_3$ to avoid discharging water of too high temperature to the tank when the main engines have been stopped for some time.

By having the mixture of air and water drawn to the coldest part of the condenser, a procedure usual with surface condensers, the proportion of air is increased and the air pump is made more efficient. An improvement which I effected enabled me to reap still greater benefit from this principle.

By injecting cold water of 18 degrees C. into the pipe which draws air from the up-flow ejector the greater part of the vapor contained therein is condensed. The relative proportion of air is increased, as it also is in an ordinary surface condenser. Consequently for a certain volume exhausted by the ejectair, the actual volume taken trom the condenser attains a very high value (see curve 4), the curve 3 gives the increase of vacuum obtained. The application of this improvement to the previous condenser on which I tried the ejectair had this curious result: with a smaller air supply, the actual volume extracted from the condenser became very great (see curve 4) and the velocity of the mixture extracted became so high in the suction pipe that cooling water returned to the up-flow ejector.

In order to continue my experiments, I had to fix a limit of $100^{m3}$ (3,531 cubic feet) of air per hour to the volume of the mixture admitted in the up-flow ejector, as I could not increase the diameter of the air suction on the condenser. For that reason I have interposed a jet pipe between the air suction and the ejector.

The reduction of the mixed exhaust was followed by an increase of vacuum, in the down-flow of this jet pipe (about $745^{mm} = 29\frac{1}{4}$ inches) such that the up-flow ejector gave $230^{m3}$ = 8,122.4 cubic feet air per hour.

The ejectair requires no special humoring; it is started simply by opening the valves $r_1$ and $r_2$ (Fig. 1), which pass the vapor to the ejectors. It is not susceptible of failure.

In getting under way it permits of establishing the vacuum before all the other auxiliaries of the main engine are put in motion, which is often of considerable advantage.

No attention is required when the apparatus is working, as it is constructed so that variations of pressure amounting to 20 per cent. are met without an appreciable effect upon the vacuum.

The upkeep and wear are practically *nil*, owing to the absence of moving mechanisms. The apparatus is very light. It seems that it would fulfill all the demands required in a marine installation.

## THE CUMMINGS ENGINE-LOG SYSTEM.

Counters and indicators showing the speed of ship and revolutions of engine continuously have been in the course of development for a number of years, with more or less success. Tachometers have been used to a considerable extent as well as recording logs. The need for satisfactory instruments of this character is greatly felt, especially when maneuvering at high speeds or where there are frequent evolutions and changes in formation.

In turbine installations it is especially desirable for the throttle man to tell immediately and accurately just what each shaft is doing in order that the exact number of revolutions signalled may be secured as quickly as possible. This is particularly useful when working engines to bells, at which times the receiver pressures serve but very little as an accurate guide to indicate the revolutions of the propeller.

A complete system for furnishing accurate indications of revolutions of shafts and speed of vessel for both engine room and conning station has been developed by the Cummings Ship Instrument Company. The following description of the complete lay out of the system is a matter of special interest to naval engineering.

Special counters of the type described are being used on official trials of recent naval vessels, and actual experience with the system in a less improved form than that now described has been generally satisfactory. The following description has been supplied by the manufacturers.—H. C. D.

### THE CUMMINGS ENGINE-LOG SYSTEM.

In both maneuvering and cruising of naval vessels the primary requisite of keeping position can be attained best when exact knowledge is had of the speed of the ship. Taking

advantage of the uniform slip of propellers at any given speed, Henry H. Cummings has developed instruments to transfer automatically the revolutions of the propellers into distance traveled through the water. Such an installation has been fitted to the latest United States battleships and auxiliary vessels.

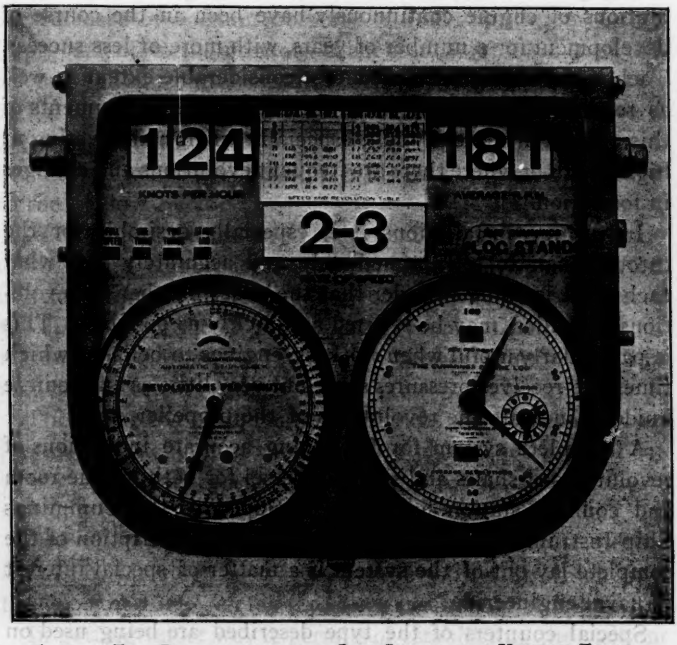



FIG. 1.—THE READINGS OF THE LOG STAND ARE VISIBLE FROM A DISTANCE ON THE BRIDGE.

This automatic method of indicating the speed and the revolutions promotes harmony between the engine-room and the bridge, and does away with a great deal of trouble and with time-consuming computations. It is no longer necessary to telephone down to the engine room for the revolutions of the engines, wait for the figure to be obtained, add up the totals for the several shafts and divide by the number of shafts, then

divide again by the number of revolutions per mile to obtain the distance traveled, and then compute the speed. The distance is now worked out accurately and automatically every 100 turns of the main engines and the revolutions per minute are given every 200 turns.

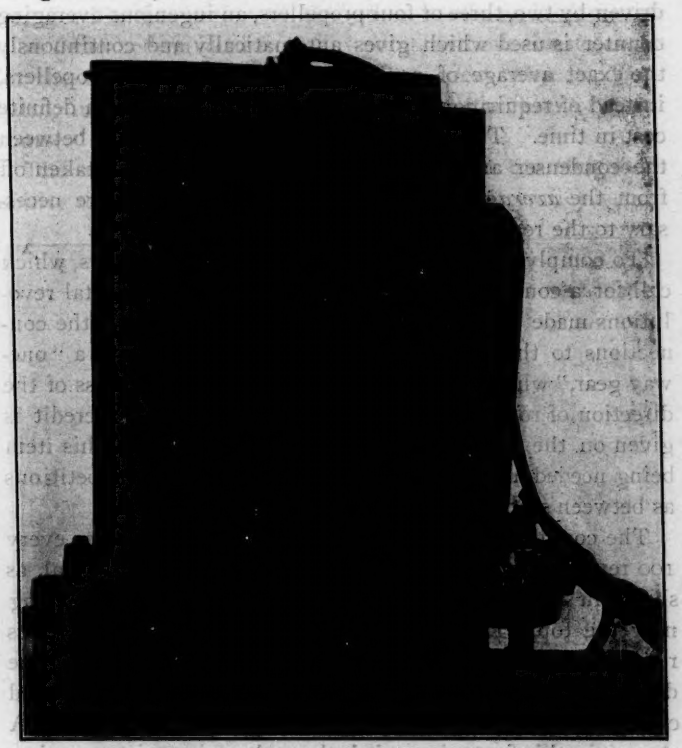


FIG. 2.—THE TWO-SHAFT AVERAGING COUNTER, LOCATED IN THE ENGINE ROOM, WITH THE ELECTRICALLY-CONTROLLED "COURSE COUNTER" ABOVE IT.

The system is operated by the vacuum in the condensers in such a way that connection between the condenser and the instruments, in the chart house or conning tower, or on the bridge, is made every 100 revolutions of the main engines. The indicator on the stop clock travels continuously during

the period between successive impulses and covers, therefore, the record of 100 revolutions. This is translated at once into revolutions per minute by means of the scale of reciprocals on the dial.

In order to average up the revolutions of the shafts of ships driven by two, three or four propellers, an ingenious averaging counter is used which gives automatically and continuously the exact average of revolutions for the various propellers, instead of requiring this to be done manually, and at a definite cost in time. The valve controlling the connection between the condenser and the instruments on the bridge is taken off from the *averaged* results, so that no corrections are necessary to the readings shown.

To comply at the same time with naval requirements, which call for a continuous rotary counter, showing the total revolutions made by the engines, both ahead and astern, the connections to the averaging counter are provided with a "one-way gear," which runs the counter forwards, regardless of the direction of rotation of the shafts. In this way full credit is given on the counter for all motion of the engines, this item being needed in connection with the engineering competitions as between ships.

The counter in the engine log moves up one unit for every 100 revolutions of the engines. Two ciphers on the dial, as shown in Fig. 3, complete the reading. The little opening near the top, "Revolutions per knot," shows the revolutions required to make one nautical mile through the water at the desired speed. This figure may be changed* as the speed changes, by means of the knurled knob at the right. A change in this figure is carried, through an ingenious gearing device, to the clockwork operating the three hands, so that no matter what may be the number of revolutions required to make one nautical mile, the hands will travel a distance so proportioned to the revolutions, as shown near the bottom of the dial, that the distance covered in nautical miles and tenths of miles will be given accurately.

*The change is very slight, except at extreme speeds, for the percentage slip of the propellers is nearly uniform at moderate speeds.

The black hand, which in Fig. 3 points to 79, is carried by friction upon the shaft of the main hand. This black hand, which is labeled "course," may be set at zero upon passing any buoy or mark, and will thus register the distance covered from any desired point. It may be set without interfering with the motion of the other hands, and, of course, shows distance traveled from any point, without the necessity for the usual calculations. This is of great value for navigating in a fog, or for covering any given distance on a certain course.

The accuracy of this instrument (based upon the navigator's ability to estimate the revolutions required per mile) has, in

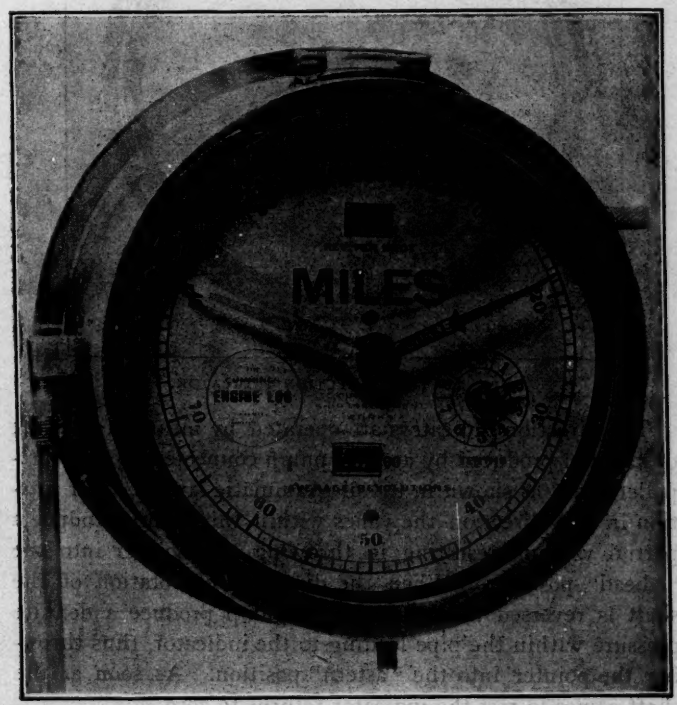


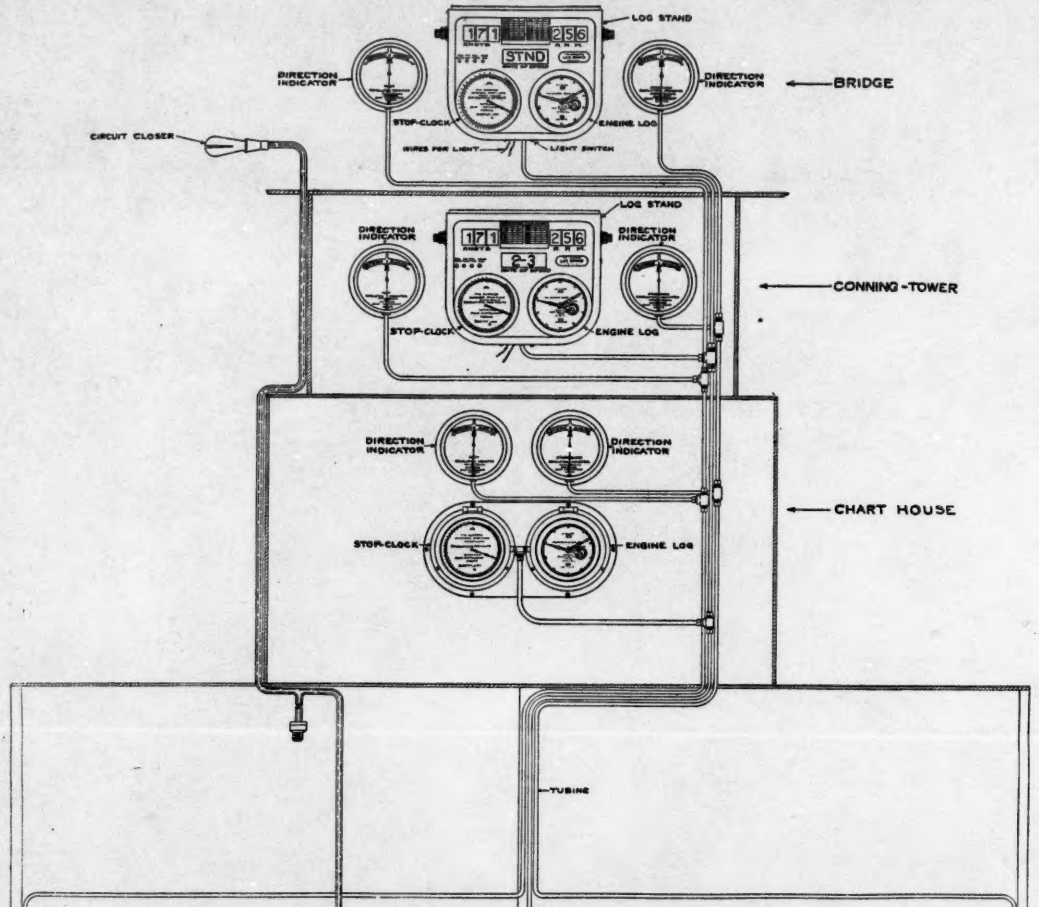

FIG. 3.—THE ENGINE LOG IS BOTH A REVOLUTION COUNTER AND A DISTANCE INDICATOR.

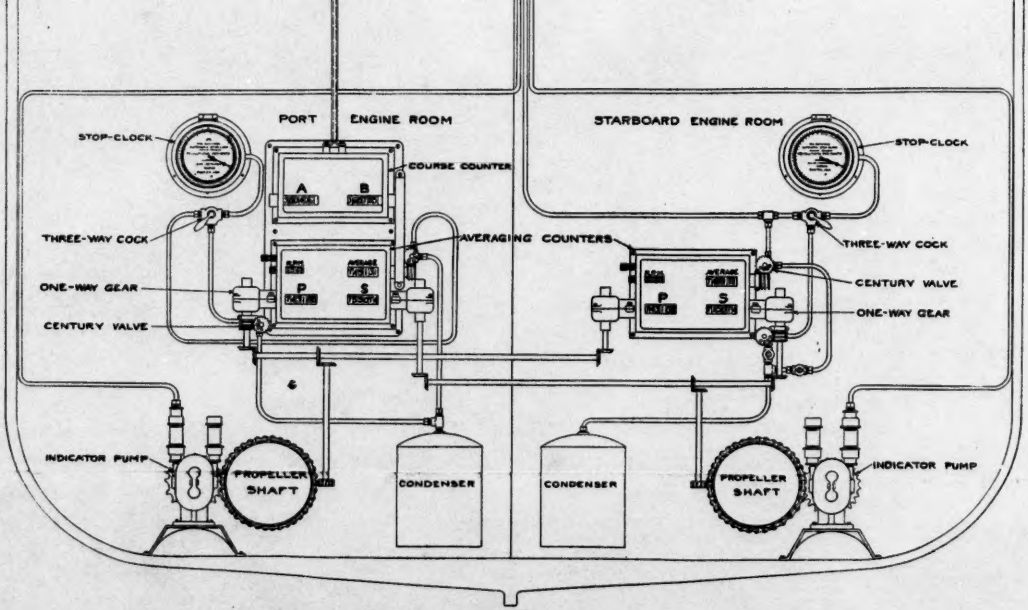
actual tests, been found surprising. Shop tests at all different speeds, and lasting for nearly two months, failed to show any appreciable error. This is due, perhaps, to the very little work required of the parts, and to their slow movement and accurate register.

FIG. 4.—THE DIRECTION INDICATOR.

The direction indicators are operated by means of pressure or vacuum produced by a small pump connected to each propeller shaft, as shown in the diagrammatic layout. For rotation in one direction the vanes within this pump produce a partial vacuum resulting in throwing the pointer into the "ahead" position. When the direction of rotation of the shaft is reversed the vanes of the pump produce a definite pressure within the pipe leading to the indicator, thus throwing the pointer into the "astern" position. As soon as the shaft comes to rest the indicator returns to zero.

The mechanism of the stop clock, Fig. 5, is similar to that of


LOG STAND
DIRECTION INDICATOR
DIRECTION INDICATOR
BRIDGE
STOP-CLOCK
ENGINE LOG
CIRCUIT CLOSER
WIRES FOR LIGHT
LIGHT SWITCH
LOG STAND
DIRECTION INDICATOR
DIRECTION INDICATOR
CONNING-TOWER
STOP-CLOCK
ENGINE LOG
DIRECTION INDICATOR
DIRECTION INDICATOR
CHART HOUSE
STOP-CLOCK
ENGINE LOG
TUBINE




This diagrammatic layout of the instruments indicates their relation to the propeller shafts and condensers, and shows a type plan of the system as applied to battleships and armored cruisers. Other vessels may be fitted with such portions of the system as requirements demand.

By manipulating the 3-way cock the revolutions per minute of each shaft or of the average of shafts will be indicated, at will of operator, on the stop clock.

an ordinary stop watch.* The vacuum in the connecting pipe drops the piston, thereby starting the hand. The vacuum in the pipe is broken when the rotary valve shuts off the condenser connection, air then being admitted to the pipe, and the piston returned by a spring. As soon as the engines make 100 turns from time of starting the clock the pipe again opens to the condenser, the piston descends and stops the clock.

FIG. 5.—THE STOP CLOCK, RUGGED AND RELIABLE, IS LOCATED IN ENGINE ROOM, CHART HOUSE AND CONNING TOWER AND ON THE BRIDGE.

The clock registers directly the number of revolutions per minute. The hand remains stationary for several seconds at the point where it was stopped, before being released and returned automatically to zero. The cycle is repeated every 200 revolutions, 100 being consumed in the motion of the

* A stop watch was used at first, but was found so unreliable, because of continually getting out of order, that a stop clock was developed heavy enough to be dependable for continuous service under all conditions.

hand and 100 in its reading and recovery. If the engines are making 200 revolutions per minute there will be an indication of the revolutions per minute once a minute. Stop clocks are usually fitted in both engine rooms, in addition to those in the chart house and elsewhere.

An auxiliary dial around the circumference of the main dial on the stop clock is graduated in knots speeds, to fit each particular ship. This can be altered when necessary, as, for instance, when the ship receives a new set of propellers, or some other change requires it. Another dial may be had for standardization purposes which carries, not the number of revolutions per minute, but the number of seconds and fifths of seconds, as in the ordinary stop watch. This dial and the one shown in the cut may be interchanged whenever it may be desirable to do so. An index may be clamped at the desired revolutions, indicating at a glance how nearly the engines are realizing what is required of them.

The two-shaft averaging counter, Fig. 2, contains three sets of counting wheels, one for the starboard revolutions, one for the port, and one for the average of the two. These are connected to the main shafts through a combination of gears and ratchets which drive the registering shaft always in one direction, regardless of the direction of rotation of the propeller shaft. By the introduction of this one-way gear much troublesome reciprocating motion in other types of counters has been eliminated. The use of gears in the averaging instruments does away with the errors inseparable from other methods of driving these counters.

In addition to the averaging counters this instrument, which is in the engine room, carries a dial for the revolutions per minute, operated by means of the knurled handle shown at the left. By keeping the teeth in contact for thirty seconds by the watch the record of revolutions per minute is shown. If more accurate results are required, or it is desired to reduce the personal error in reading the watch, another gear may be thrown in and the reading taken for five minutes, giving the average revolutions per minute over that period.

This device permits the engineers to check up the revolutions per minute, independently of the instruments on the bridge.

The auxiliary (Smith-Cummings) counter above the averaging counter, designed particularly for standardization of the propellers and for trial runs, contains two counting cylinders, A and B. These are operated individually from the averaging counter below, one cylinder running while the other is stationary. This shows accurately the total "averaged" number of

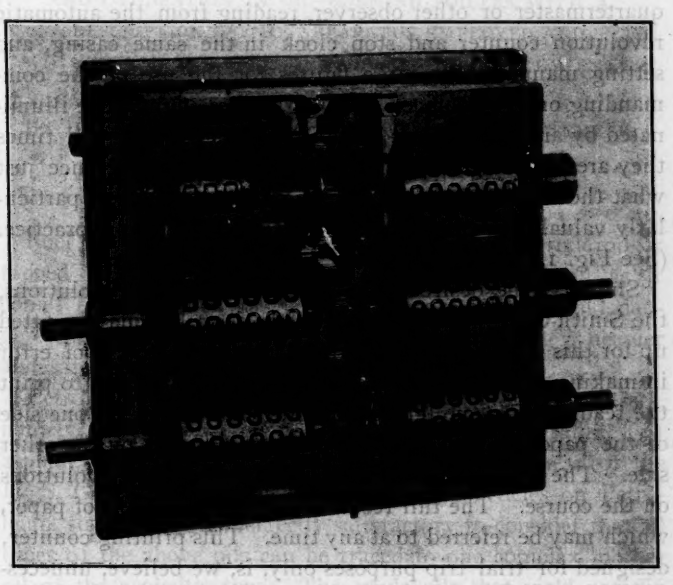


FIG. 6.—THE FOUR-SHAFT AVERAGING COUNTER OPERATES BY MEANS OF GEARS, ON THE SAME PRINCIPLE AS THE TWO-SHAFT COUNTER.

revolutions for a given time or over a given distance. It is controlled electrically from the bridge or any other desired point. The observer on deck closes the circuit when the line is crossed, thus immediately starting the stationary counter and stopping the one which was running. When the circuit is again closed these operations are repeated. This instrument, especially useful in determining the revolutions for each period during a long run, enables the officer in

charge to obtain an extremely accurate reading of the revolutions per minute, averaged over the period. It is fitted usually in one engine room only.

On the bridge and in the conning tower of the latest ships are located indicator boxes for showing both the revolutions per minute and the speed of the ship in large figures visible from any position or distance at which the officer on duty may be. These "distance" indications are controlled by a quartermaster or other observer, reading from the automatic revolution counter and stop clock in the same casing, and setting manually the large figures for the use of the commanding or other officer. At night these figures are illuminated by an incandescent light in the case, and at all times they are so arranged that the officer can see at a glance just what the engines and the ship are doing. These are particularly valuable in maneuvering vessels and in target practice. (See Fig. 1.)

Should it be desired to print the readings of revolutions, the Smith-Cummings counter already described may be fitted up for this purpose. This eliminates the possibility of error in making a reading. It has been found convenient to print the reading of a counter before going on a course on one side of the paper, and the reading after coming off on the other side. The difference between the two will be the revolutions on the course. The full record is thus on one piece of paper, which may be referred to at any time. This printing counter, designed for trial trip purposes only, is, we believe, unnecessary for general service use.

We have spoken above mostly of twin-screw ships, showing the averages of port and starboard propellers. For quadruple-screw ships the same arrangement obtains except that there are more gears involved. One set averages the two shafts on the port side of the ship; another averages the two shafts on the starboard; the third averages these two averages. This device is illustrated. The arrangement for averaging the revolutions on a triple-screw ship is somewhat different in detail, but operates on the same general principles and gives the same dependable results.

# OIL BURNING.

## FUEL OILS, THEIR COMPOSITION, GENERAL CHARACTERISTICS, AND UTILIZATION IN OIL-BURNING BOILERS WITH A DESCRIPTION OF THE APPARATUS EMPLOYED AND THE MEANS USED TO PROPERLY BURN THEM.

By Lieut. Com. John J. Hyland, U. S. N., Member.

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Fuel oil may, in general, be classed as topped crude oil or topped petroleum. That is, petroleum or crude oil as it comes from the well has its lighter, more valuable and dangerous constituents removed by distillation, leaving a heavier and safer oil for oil-burning purposes.

Petroleum or crude oil may be divided into three classes, namely, those which on distillation yield (1) paraffin, (2) asphalt, and (3) olefin. Doctor David T. Day, head of the U. S. Geological Survey, in his very able articles on Petroleum, states that it is perfectly satisfactory to consider that all types of the above oils can be traced to one complex mother oil of the asphaltic type. The crude oils that yield paraffin are generally found in the Appalachian range and the middle west. Those that yield asphalt are generally found in Mexico, California and the Gulf Coast. Those that yield olefin are generally found in the oil fields of Russia.

Crude petroleum, as it comes from the ground, is a complex mixture of many hydrocarbons, and it varies greatly in both its chemical and its physical properties, depending on its locality. Doctor Day states that the yield of the United States is two hundred and twenty (220) million barrels of

petroleum, and that the division of the above by distillation or fractionation is as follows:

20 million barrels of Gasoline.
65 million barrels of Kerosene.
20 million barrels of Gas Oil.
20 million barrels of Lubricating Oil.
80 million barrels of Fuel Oil.
15 million barrels of Paraffin, Coke and Loss.
220 million barrels total.

The following table prepared by Doctor Day shows the world's production of petroleum in barrels for the years 1907 to 1911 inclusive. Since that time the output of the Mexican fields has greatly increased due to the discovery of new fields near Tampico and Tuxpam.

| Country. | 1907. | 1908. | 1909. | 1910. | 1911. |
|---|---|---|---|---|---|
| United States | 166,095,335 | 178,527,355 | 183,170,874 | 209,557,248 | 220,449,391 |
| Russia | 61,850,734 | 62,186,447 | 65,970,350 | 70,336,574 | 66,183,691 |
| Mexico | 1,000,000 | 3,481,410 | 2,488,742 | 3,332,807 | 14,051,643 |
| Dutch East Indies | 9,982,597 | 10,283,357 | 11,041,852 | 11,030,620 | 12,172,949 |
| Roumania | 8,118,207 | 8,252,157 | 9,327,278 | 9,723,806 | 11,101,878 |
| Galicia, A. H. | 8,455,841 | 12,612,295 | 14,932,799 | 12,673,688 | 10,485,726 |
| India | 4,344,162 | 5,047,038 | 6,676,517 | 6,137,990 | 6,451,203 |
| Japan | 2,010,639 | 2,070,145 | 1,889,563 | 1,930,661 | 1,658,903 |
| Peru | 756,226 | 1,011,180 | 1,316,118 | 1,330,105 | 1,398,036 |
| Germany | 756,631 | 1,009,278 | 1,018,837 | 1,032,522 | 955,764 |
| Canada | 788,872 | 527,987 | 420,755 | 315,895 | 291,096 |
| Italy | 59,875 | 50,966 | 42,388 | 42,388 | *71,905 |
| Others | *30,000 | *30,000 | *30,000 | *30,000 | *200,000 |
| Total | 264,249,119 | 285,089,615 | 298,326,073 | 327,474,304 | 345,512,185 |

*Estimated.

The following tables prepared from various sources, and that on different oils from the United States prepared by the writer, give in general terms the various chemical constituents and characteristics of the various fuel oils.

| Kind of oil. | Gravity. | Flash point P.M. | B.t.u. per gallon or pound. | Carbon. | Hydrogen. | Nitrogen. | Sulphur. | Oxygen, etc. | Viscosity at 70 degrees F. Engler. |
|---|---|---|---|---|---|---|---|---|---|
| | °B. | °F. | | | | | | | |
| Texas Oil Co | 24 | 210 | 145,000 | 86.40 | 12.25 | 0.58 | 0.74 | 0.03 | From 5 to 10. |
| Texas Co., Mexican | 20 | 65 | 143,720 | 84.40 | 11.80 | 0.50 | 2.80 | 0.50 | (?) |
| California oil | 16.3 | 240 | 157,740 | 86.93 | 11.66 | 0.75 | 0.48 | 0.18 | (?) |
| Texas Co., Mexican | 17.3 | 126 | 150,250 | 84.00 | 11.70 | 0.72 | 3.14 | 0.44 | 116.1 at 100° F. |
| Texas Co. oil | 23.6 | 207 | 147,600 | 86.20 | 12.70 | 0.25 | 0.50 | 0.35 | .5 |
| Standard oil, Mexican | 17.1 | 145 | 150,250 | 83.50 | 11.35 | 0.42 | 3.06 | 1.67 | 300.5 |
| Standard oil, Mexican | 15.4 | 202 | 151,390 | 83.90 | 11.55 | 0.42 | 3.47 | 0.66 | 715 |
| Illinois oil | 27.2 | 146 | 145,630 | 84.60 | 13.05 | 0.27 | 0.30 | 1.78 | 6.8 |
| Indiana oil | 29.6 | 144 | 144,250 | 84.20 | 13.65 | 0.15 | 0.50 | 1.50 | 4.83 |
| Louisiana oil | 19.8 | 275 | 147,160 | 86.30 | 12.30 | 0.14 | 0.43 | 0.83 | 50.2 |
| "Star," California oil | 23.9 | 180 | 146,070 | 86.40 | 12.20 | 0.33 | 0.42 | 0.65 | 2.96 |
| "Richmond," Cal. oil | 17.1 | 228 | 151,210 | 86.60 | 11.70 | 0.58 | 0.70 | 0.42 | 82.2 |
| Lima, Ohio | 30.4 | 149 | 142,660 | 85.20 | 12.80 | 0.12 | 0.40 | 1.48 | 7.0 |
| Mexican gas oil | 34.2 | 151 | 139,760 | 84.70 | 14.00 | 0.16 | 1.13 | 0.01 | 1.56 |
| Mexican residue | 10 | 374 | 153,910 | 83.60 | 12.00 | 0.45 | 3.86 | 0.09 | 67 at 200° F. |
| Gulf Ref. Co. oil | 29.2 | 170 | 144,015 | 85.00 | 13.50 | 0.10 | 0.20 | 1.20 | 10 |
| Union Oil Co., Cal | 12.9 | 285 | 151,400 | 84.60 | 13.40 | 0.68 | 0.93 | 0.39 | 102 at 125° F. |
| Union Oil Co., Cal | 13.2 | 262 | 152,465 | 85.00 | 12.50 | 0.77 | 0.83 | 0.90 | 80 at 125° F. |
| Union Oil Co., Cal | 12.9 | 280 | 151,245 | 84.50 | 13.00 | 0.77 | 0.87 | 0.86 | 96 at 125° F. |
| Union Oil Co., residuum | 18.5 | 223 | 150,100 | 86.20 | 12.00 | 1.03 | 0.74 | 0.03 | 93.5 |
| Producers' crude, Union Oil Co | 16.1 | 174 | 152,630 | 85.10 | 13.00 | 0.66 | 0.70 | 0.54 | 98.0 |
| Coalinga Field | 16.5 | 186 | 151,260 | 86.20 | 12.60 | 0.52 | 0.59 | 0.09 | 73.5 |
| Toltec oil | 11.7 | 220 | 149,330 | 83.80 | 11.20 | 0.43 | 4.50 | 0.07 | 166 at 125° F. |
| Avon-California | 17.1 | 168 | 148,500 | 85.50 | 12.10 | 0.66 | 0.78 | 0.96 | 65.2 |
| Gaviota, California | 17.1 | 230 | 147,745 | 83.00 | 11.50 | 0.59 | 2.51 | 2.40 | 89.2 at 100° F. |
| Anglo-Mexican Co., No. 1. | 15.8 | 188 | 148,500 | 83.80 | 11.75 | 0.37 | 3.18 | 0.90 | 47.6 at 100° F. |
| Anglo-Mexican Co., No. 2. | 16.2 | 238 | 150,500 | 84.00 | 12.00 | 0.36 | 2.78 | 0.86 | 71.54 at 100° F. |
| Anglo-Mexican Co., No. 3. | 17.3 | 164 | 148,850 | 83.40 | 11.95 | 0.34 | 3.40 | 0.91 | 96.38 at 100° F. |
| Camden Co. coal tar by product | 6 | 108 | 158,120 | 90.90 | 8.00 | 0.14 | 0.46 | 0.50 | 2.3 |
| Mexican oil | 17.6 | 182 | 147,500 | 83.60 | 11.75 | 0.32 | 2.70 | 1.63 | 49.24 at 100° F. |
| Toltec or Panuco oil | 12 | 124 | 149,500 | 82.50 | 10.95 | 0.45 | 4.53 | 1.57 | 375 at 105° F. |

REPORT OF BUREAU OF MINES.

| Field. | Specific gravity. | B.T.U. per gal. | Flash, open cup. | Viscosity at 60° F., Engler. | Carbon. | Hydrogen. | Nitrogen. | Sulphur. | Undetermined. |
|---|---|---|---|---|---|---|---|---|---|
| | °B. | | °C. | | | | | | |
| Kern River | 15.16 | 148,980 | 108 | 915.0 | 86.36 | 11.27 | 0.74 | 0.89 | 0.74 |
| | 14.78 | 149,610 | 102 | 690.0 | | | | | |
| Coalinga | 17.52 | 148,130 | 88 | 341.5 | 86.37 | 11.30 | 1.14 | 0.60 | 0.59 |
| | 17.29 | 148,262 | 72 | 64.5 | | | | | |
| McKittrick | 16.37 | 148,276 | 87 | 200.0 | 86.51 | 11.41 | 0.58 | 0.74 | 0.76 |
| | 15.83 | 146,680 | 74 | 160.7 | | | | | |
| Midway | 16.34 | 148,345 | 78 | 518.1 | 86.58 | 11.61 | 0.74 | 0.82 | 0.25 |
| | 16.14 | 148,149 | 61 | 137.9 | | | | | |
| Sunset | 14.37 | 149,302 | 89 | 527.2 | 85.64 | 11.37 | 0.84 | 1.06 | 1.09 |
| | 14.26 | 149,010 | 71 | 604.2 | | | | | |

LIQUID-FUEL BOARD REPORT.

| Kind of Oil. | Specific gravity. | Flash point. | Fire.—Vaporization point. | B.T.U. per lb. | Carbon. | Hydrogen. | Sulphur. | Oxygen, N. & O. |
|---|---|---|---|---|---|---|---|---|
| | | °F. | °F. | | | | | |
| Beaumont crude | 0.924 | 180 | 200 | 19,060 | 84 60 | 10.90 | 1.63 | 2.87 |
| Beaumont refined | 0.926 | 216 | 240 | 19,481 | 83.26 | 12.14 | 0.50 | 3.83 |
| California | .966 | 311 | 230 | ... | 81.52 | 11.01 | 0.55 | 6.92 |

CALORIFIC POWER OF VARIOUS DESCRIPTIONS OF PETROLEUM, ETC.
(Taken from Redwood.)

| | Specific gravity at 0°C. | Chemical composition. | | | Effects in heat units. |
|---|---|---|---|---|---|
| | | Carbon. | Hydrogen. | Oxygen. | |
| | | | | | Calories |
| Heavy petroleum from West Virginia | 0.873 | 83.5 | 13.3 | 3.2 | 10,180 |
| Light petroleum from West Virginia | 0.8412 | 84.3 | 14.1 | 1.6 | 10,223 |
| Light petroleum from Pennsylvania | 0.816 | 82.0 | 14.8 | 3.2 | 9,963 |
| Heavy petroleum from Pennsylvania | 0.886 | 84.9 | 13.7 | 1.4 | 10,672 |
| American petroleum | 0.820 | 83.4 | 14.7 | 1.9 | 9,771 |
| Petroleum from Parma | 0.786 | 84.0 | 13.4 | 1.8 | 10,121 |
| Pechelbronn | 0.912 | 86.9 | 11.8 | 1.3 | 9,708 |
| Pechelbronn | 0.892 | 85.7 | 12.0 | 2.3 | 10,020 |
| Schwabweiler | 0.861 | 86.2 | 13.3 | 0.5 | 10,458 |
| Schwabweiler | 0.829 | 79.5 | 13.6 | 6.9 | ... |
| Hanover, Eddese | 0.892 | 80.4 | 12.7 | 6.9 | ... |
| Hanover, Wietze | 0.955 | 86.2 | 11.4 | 2.4 | ... |
| East Galicia | 0.870 | 82.2 | 12.1 | 5.7 | 10,005 |
| West Galicia | 0.885 | 85.3 | 12.6 | 2.1(NO) | 10,231 |
| Shale oil from Ardeche, Vagnas | 0.911 | 80.3 | 11.5 | 8.2(OSN) | 9,046 |
| Coal tar from Paris gas works | 1.044 | 82.0 | 7.6 | 10.4 | 8,916 |
| Petroleum from Balakhani | 0.822 | 87.4 | 12.5 | 0.1 | 11,700 |
| Light petroleum from Baku | 0.884 | 86.3 | 13.6 | 0.1 | 11,460 |
| Heavy petroleum from Baku | 0.938 | 86.6 | 12.3 | 1.1 | 10,800 |
| Petroleum residue from Baku factories | 0.928 | 87.1 | 11.7 | 1.2 | 10,700 |
| Petroleum from Java | 0.923 | 87.1 | 12.0 | 0.9 | 10,831 |
| Heavy oil of pine (Landes) | 0.985 | 87.7 | 10.4 | 2.5 | 10,081 |

FUEL OILS. (Taken from Lewes.)

| | Carbon. | Hydrogen. | Oxygen, etc. |
|---|---|---|---|
| American | 84.9 | 13.7 | 1.0 |
| Russian | 86.6 | 12.3 | 1.1 |
| Borneo | 87.8 | 10.78 | 1.24 |
| Texas | 85.6 | 11.03 | 3.51 |
| Caucasus | 84.9 | 13.96 | 1.25 |
| Burmah | 86.4 | 12.10 | 1.50 |

LIQUID FUELS. (Taken from Lewes.)

| | Specific gravity. | Flash point. | Calorific value by bomb. | |
|---|---|---|---|---|
| | | | Calories. | B.t.u.'s. |
| | | ° F. | | |
| American residuum | .886 | 350 | 10,904 | 19,627 |
| Russian Ostatki | .956 | 308 | 10,800 | 19,440 |
| Texas | .945 | 244 | 10,700 | 19,242 |
| Burmah | .920 | 230 | 10,480 | 18,864 |
| Barbadoes | .958 | 210 | 9,899 | 17,718 |
| Borneo | .936 | 285 | 10,461 | 18,831 |
| Shale oil | .875 | 288 | 10,120 | 18,217 |
| Blast-furnace oil | .979 | 206 | 8,933 | 16,080 |
| Heavy tar oil | 1.084 | 218 | 8,916 | 16,050 |

An examination of the above tables shows that these oils may be divided into certain groups which contain approximately the same percentage of chemical constituents and approximately the same heating value per gallon.

However, as will be shown later, these characteristics have in general very little to do with their proper burning. Before an oil is burned certain tests should be made to determine its characteristics, and some of these are required to be made before an oil is accepted. These tests are (1) determination of the gravity of the oil, (2) determination of the flash point, (3) determination of the water and sediment, (4) determination of the calorific value, (5) determination of the sulphur contents, (6) determination of the ultimate analysis of the oil, and, lastly, (7) the determination of its viscosity at various temperatures.

(*a*) The gravity of an oil is found by filtering a sample of the oil into a 100 c.c. cylinder and allowing a hydrometer to float freely in this oil. The hydrometer has a scale graduated in degrees Beaumé on its face, and also a temperature scale in degrees Fahrenheit. Both indications are noted, and by means of the tables supplied by Tagliabue the gravity at 60 degrees F. can easily be found by interpolation, in case the temperature of the oil is not 60 degrees F. The specific gravity of the oil can be found from the formula,

$$\text{Specific gravity} = \frac{140}{130 + \text{° Beaumé}}.$$

The tables prepared by the Inspector of Petroleum of the New York Produce Exchange Annex give the specific gravity of light oils for every 1/10 of a degree on the Beaumé scale.

(*b*) The flash point of an oil is the temperature to which an oil must be heated to give off vapors which, when mixed with air, form an explosive mixture. This test is usually found either by the open-cup method or the closed-cup method. The Navy Department, and general practice usually give the flash by the Abel or Pensky-Marten closed-cup method. In the Pensky-Marten closed cup tester the oil is first filtered to prevent foaming, in case any water is present. It is then placed in a brass cup of standard size and filled to a mark on the inside of the cup. This cup rests in an outer vessel which forms a water jacket, the outer vessel being supported by three legs and heated by an alcohol burner placed below. The oil cup is fitted with a tightly fitting cover which carries a thermometer and allows it to extend into the liquid, a stirrer for agitating the oil, made of two paddles, and a small testing flame or wick. The cover has triangular holes cut in it, and these holes are covered by a circular plate which is rotated to expose the opening and at the same time actuates the wick which dips into the opening. In case the temperature of the oil is not near the flash the wick will be extinguished when dipped. As the flash point is neared the wick flame increases in brightness, and at the actual flash a bluish flame shoots up through the opening with a perceptible flash.

The fire point or burning point of an oil is the temperature at which vapors are given off which, when ignited, burn continuously. The fire test is made in an open tester. The fire point is usually about 25 degrees F. above the flash.

(*c*) The test for water and sediment is now usually made with benzol as follows: 50 c.c. of the oil and 50 c.c. of benzol are placed in a high-speed centrifuge and revolved rapidly for some time and then the percentage of water is read off in terms of the total 100 per cent. In case the sample contains a large percentage of water the test is made by the distillation method.

(*d*) The calorific value of an oil is usually determined by the bomb calorimeter, Mahler's bomb calorimeter or any other standard make being used. Approximate formulas for determining the B.t.u.'s without a bomb calorimeter are as follows:

B.t.u.'s = 18,650 + 40 (Beaumé reading —10). "Sherman & Kropff."

B.t.u.'s = 17,680 + 60 (Beaumé reading). "J. N. LeConte."

(*e*) The sulphur contents of an oil may be determined by any of the following methods:

1. Dry fusion with alkalies and subsequent oxidation with bromine.
2. Dry fusion with a mixture of alkalies and oxidizing agents.
3. Treatment with wet alkalies and oxidizing agents.
4. Oxidizing with fuming nitric acid at high pressures.
5. Burning in pure oxygen at atmospheric pressure.
6. Burning in a stream of pure oxygen.
7. Burning in a lamp in atmospheric oxygen.
8. Burning in a bomb calorimeter with pure oxygen under a pressure of 30 to 40 atmospheres.

The last method is the best and most reliable and is the one usually used.

(*f*) The ultimate analysis of an oil is made by a chemist, the following being the constituents found: Carbon, hydrogen, nitrogen, sulphur, oxygen, water and sediment.

(*g*) The viscosity of an oil is its rate of flow, of a certain amount, through a specific orifice at a certain temperature. There are various makes of viscosimeters used, namely, Engler's, Saybolt's, Redwood's, Kunkler's and a great many others, each of which use a different liquid and a different volume as a standard. The Engler, which is the standard in the U. S. Navy, puts 240 c.c. in the cup and allows 200 c.c. to run out, the comparison being water at 70 degrees F. as unity. The Saybolt puts 70 c.c. in the cup and allows 60 c.c. to run out, a stop watch timing the interval, and the second is the standard.

It is pretty well established from recent investigations of different fuel oils at the Fuel-Oil Testing Plant by Prof.

Hulett and Mr. Burrell, of the Bureau of Mines, and Doctor Day, of the Geological Survey, that it is impossible to get an explosive mixture from the gases given off by oils when heated and mixed in combination with air below the flash point of the oil.

It was found that in order to have an explosive mixture 1 per cent. to 1.5 per cent. of vapor is needed. The table following gives this data.

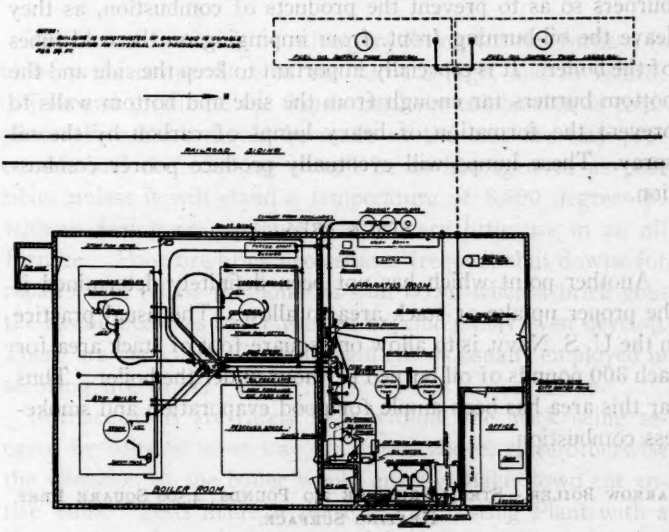
Another fact disclosed is that where oil is stowed in air-tight tanks, so that no oxygen is present, the gases are absolutely non-explosive because they cannot receive oxygen to support combustion.

VAPOR TABLE.

| Kind of oil. | Temperature to which oil was heated. | | Oxygen, per cent. | Vapor, per cent. | Flash point. | |
|---|---|---|---|---|---|---|
| | Degs. C. | Degs. F. | | | Degs C. | Degs. F. |
| Mexican crude, 17.3, Beaumé..... | 53 | 127 | 20.4 | 1.00 | 52 | 126 |
| | 73 | 163 | 20.0 | 1.10 | ... | ... |
| Mexican gas oil..... | 23 | 73 | 20.8 | .20 | 64 | 147 |
| | 63 | 145 | 20.1 | .90 | ... | ... |
| | 92 | 198 | ... | 1.40 | ... | ... |
| Lima oil..... ..... | 24 | 75 | 20.9 | .20 | 55 | 131 |
| | 55 | 131 | 20.1 | 1.20 | ... | ... |
| | 77 | 171 | 18.1 | ... | ... | ... |
| Mexican crude, 17, Beaumé..... | 55 | 131 | 20.7 | .40 | 70 | 158 |
| | 71 | 160 | 20.4 | .80 | ... | ... |
| | 94 | 201 | 19.7 | 1.40 | ... | ... |
| Louisiana oil..... ..... ..... | 97 | 207 | 20.4 | .95 | 130 | 266 |
| | 75 | 167 | 20.6 | .70 | ... | ... |
| Mexican crude, 15.4, Beaumé..... | 72 | 162 | 20.4 | .70 | 94 | 201 |
| | 95 | 203 | 18.6 | 1.50 | ... | ... |
| Indiana oil..... | 35 | 95 | 20.5 | .30 | 54 | 147 |
| | 66 | 151 | 20.3 | .90 | ... | ... |
| | 87 | 189 | 20.0 | 1.30 | ... | ... |
| Navy contract (Oklahoma)..... | 50 | 122 | 20.3 | .50 | 75 | 167 |
| | 75 | 167 | 20.1 | 1.0 | ... | ... |
| | 95 | 203 | 19.8 | 1.40 | ... | ... |
| California Star oil..... | 50 | 122 | 19.9 | .55 | 85 | 185 |
| | 98 | 208 | 15.5 | 1.37 | ... | ... |

The foregoing facts regarding fuel oils being set forth, I will now turn to the burning of these oils in boilers and furnaces and in a general way try to describe the methods used and the general rules that should be followed to burn any oil properly.

Burning fuel oil in a boiler, under the mechanical or pressure system of atomization, may be conveniently divided into three heads: (1) The boiler furnace and its accessories; (2) The oil-burning apparatus proper, consisting of the burners, the tuyères, air cones or air registers, as they are called, and (3) The air regulation, oil regulation and aids used to make the combustion as perfect as possible.




DIAGRAMMATIC PLAN OF OIL BURNING TESTING PLANT
NAVY YARD, PHILADELPHIA, PA.

The purpose of this part of the article is to try to describe under the three heads noted above the mechanical system of burning oil, the various apparatus used since its development, and the changes made therein since the mechanical system was first successfully introduced, and briefly to describe the methods of burning oil with steam or air burners.

In order to burn fuel oil properly there are three essential conditions: (1) The gases of combustion must be kept at the ignition temperature long enough to properly burn them; (2) There must be an intimate mixing of the particles of air and oil; (3) The air supply must be sufficient for combustion.

In practice, the first condition is secured by providing proper furnace arrangements, ample combustion space and suitable refractory linings. The aim of furnace design should be to secure such shape, volume and arrangement that the combustion of the gases is completed in the boiler and that the gases are suitably retarded so that all possible heat is absorbed by the heating surface. Another essential point is to locate the burners so as to prevent the products of combustion, as they leave the oil-burning front, from impinging on the cold tubes of the boiler. It is especially important to keep the side and the bottom burners far enough from the side and bottom walls to prevent the formation of heavy lumps of carbon by the oil spray. These lumps will eventually produce poorer combustion.

## STACK AREA.

Another point which has not been definitely determined is the proper uptake or stack area to allow. The usual practice in the U. S. Navy, is to allow one square foot of stack area for each 300 pounds of oil burned per hour under the boiler. Thus far this area has been ample for good evaporation and smokeless combustion.

YARROW BOILER.—STEAM PRESSURE 240 POUNDS; 4,500 SQUARE FEET HEATING SURFACE.

| Stack area, square feet. | Pounds of oil burned per hour per square foot H.S. | Evaporation from and at 212 degrees per pound of oil. | Stack temperature, degrees F. | Per cent. $CO_2$. | Air pressure in fire-room, inches of water. | Velocity head, inches of water. | Power. |
|---|---|---|---|---|---|---|---|
| 15.90 | .9809 | 13.83 | 835 to 890 | 13.8 to 14.2 | 5.90 | 3.50 | Full. |
| 12.57 | .9750 | 13.32 | 825 to 845 | 14.0 to 14.2 | 5.50 | 3.50 to 3.20 | |
| 9.62 | .9730 | 13.68 | 800 to 805 | 14.6 to 14.8 | 5.40 | 3.45 | |
| 15.90 | .7383 | 14.212 | 730 to 740 | 13.2 to 14.0 | 4.20 to 3.50 | 2.15 to 1.91 | ¾ |
| 12.57 | .7327 | 14.38 | 690 to 725 | 14.0 to 14.4 | 3.50 | 2.00 to 2.31 | |
| 9.62 | .728 | 14.60 | 660 to 680 | 14.0 to 14.2 | 3.85 | 2.25 to 2.35 | |
| 15.90 | .4867 | 15.07 | 550 to 555 | 13.4 to 14.0 | 3.50 | 1.10 | ½ |
| 12.57 | .4690 | 15.17 | 750 to 575 | 13.0 | 3.00 | 1.15 to 1.20 | |
| 9.62 | .4693 | 15.07 | 525 to 535 | 13.6 to 14.0 | 3.30 | 1.30 to 1.40 | |

Experiments with reduced stack area have been conducted as shown in preceding table, which appear to indicate that the area can be considerably reduced from the usual practice without reducing the capacity of the boiler, and improving the combustion as indicated by the $CO_2$ content.

## FIRE BRICK.

The use of fire brick or refractory lining for the furnace is to keep the temperature of the furnace high, and to retard, diffuse and distribute the flames until the combustion is complete. Fire brick should be made of the best fire clay procurable and the crushing should be as homogenous as possible; unless it will stand a temperature of 3,200 degrees F. without fusing or cracking it is of very little use in an oil furnace. Poor fire brick necessitates frequent shut downs for repairs, and unless the boiler is shut down when a brick goes the casing becomes badly warped or a hole may even develop. There are shown in Fig. 1 three methods usually employed in securing fire brick.

Furnace walls are rarely built without the brick being secured by bolts in some way, as shown above, since otherwise the vibration of the boiler would quickly shake down the entire walls. Tests made at the Fuel-Oil Testing Plant with a Brown fixed-focus radiation pyrometer and a Féry variable-focus radiation pyrometer show that the temperature of the furnace varies from 2,500 degrees F. near the burners to 2,800 degrees F. well inside the furnace. When the oil and air are improperly mixed, due to improper regulation, the temperature at the burners dropped in many cases to 1,900 degrees F. The necessity for good regulation is shown thereby.

The Johns-Manville Co. and the Botfield Specialties Co. each put out a high-temperature cement, both for laying up the fire brick and for effecting repairs to the broken brickwork and holes found due to bricks fusing. They are generally used in the form of a light mortar, and are allowed to settle for a few hours and then hardened by lighting fires

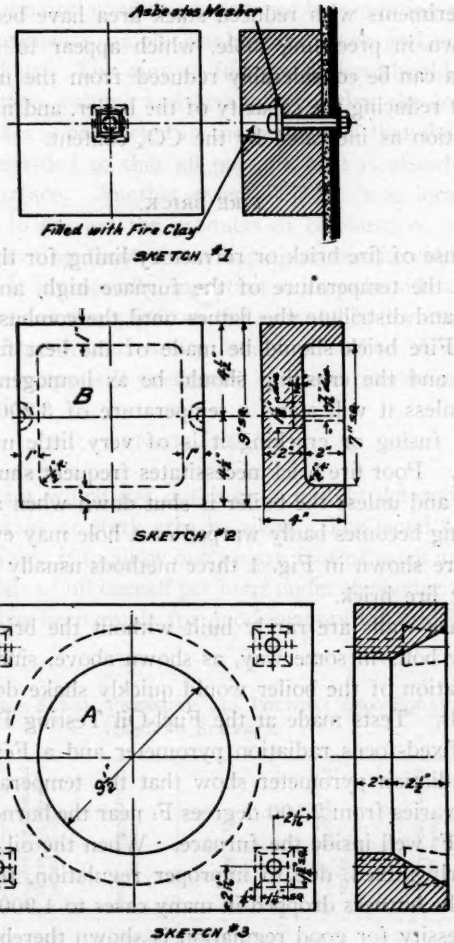



FIG. 1.

with one burner to give a gentle heat. In general, the ingredients of the Johns-Manville cement are fire clay, cement, ground glass or silicate of soda, finely ground magnesia and lamp black. A mixture of carborundum fire sand and silicate

of soda made into a heavy wash or paste and applied with a brush forms an excellent coating on the brick work, which not only prolongs its life, but materially assists in maintaining the high temperature needed in the furnace.

BURNERS.—THE PRINCIPLES EMPLOYED IN THEIR MANUFACTURE AND THE VARIOUS TYPES USED.

In oil burning the atomization of the oil by means of a mechanical burner under pressure as distinguished from vaporization or gasification in the burner is the only means which has attained any success in the Navy. Various advocates of burning oil, especially in the early days, insisted that the only way to burn oil was by some treatment of the oil which admitted it to the furnace in form of vapor. These systems, while successful in metallurgical work, are not successful in boiler work, where great capacity is needed. They also introduce a factor of danger. The mechanical burner sprays or atomizes the oil by means of pressure alone, without the use of compressed air, steam or any other exterior atomizing agent. In the U. S. Navy two methods are employed to accomplish this purpose: (1) By forcing the oil through a passage of helical form like the screw thread, and (2) By delivering the oil tangentially to a circular chamber (oil chamber) from which there is a central outlet.

Before proceeding to describe the various burners that are built on the above principles it will be well to describe the requirements of a successful burner and at the same time correct various popular fallacies regarding them. Any successful burner must reduce its atomization to very minute particles, so that, when issuing from the burner, they will not drop to the furnace bottom before combining with air; and the friction in the central or oil chamber must be very small. It is quite possible to work out the coefficients of friction of different burners. The velocity of an oil issuing from a chamber $=$ $V = \sqrt{2gh}$, where $h$ is the head due to oil pressure. The quantity Q passed through in a given time $=$ $Q = K \times v \times$

A, where A = area of the orifice and K = coefficient of friction. Q and A are easily found, and have been found at the Fuel-Oil Testing Plant and curves drawn, so it would be very easy to find KV and compare them.

The idea that oil issuing from a burner tip of the mechanical form goes out in a revolving spray is a fallacy that should be corrected. The oil issues in straight lines, in a hollow conical spray of various angles up to 90 degrees. The rotation is entirely within, and confined to the oil chamber proper. The three most familiar burners of the first type, namely, those that secure the rotating motion by means of a helical screw thread, are the Schutte-Koerting, Thornycroft, and Howden.

### THE SCHUTTE-KOERTING BURNER.

This burner is installed on the *Burrows, Mayrant, Warrington, Ammen, McCall, Patterson, Jarvis, Beale, Aylwin, Parker, Balch and Downes.* Figs. 2 and 3.

The tip of this burner is cut out to receive a small spindle, less in diameter than the oil chamber except at the end. A triple-parallel thread is cut on this spindle, which just fits the chamber and forms with it three helical passages which deliver the oil to a smaller chamber at the end of the spindle which communicates with the central outlet. The spindle is not adjustable, so the output is controlled by the temperature and pressure of the oil. The burner is fitted with a yoke and hand screw which holds it in position and provides a ready means for disconnecting it.

### THORNYCROFT BURNER.

This burner was installed on the *Roe, Terry* and *Monaghan.* Fig. 4.

In this burner the oil receives a whirling motion, passing through a spiral groove into a central chamber communicating with the outlet orifice of the tip. The tip fits on to a nozzle, in which there is a cylindrical hole about the same diameter

as the central chamber and concentric with the axis of the burner. In the surface of this cylindrical hole a thread of square section is cut, of very slight depth at the end coinciding with the central chamber in the tip, but increasing rapidly in depth towards the opposite end of the burner at which oil is admitted. A spindle fits into the cylindrical hole of the nozzle

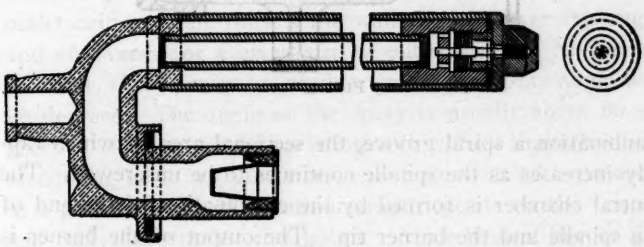


FIG. 2.

and on this spindle there is a corresponding thread, accurately fitting the threads of the nozzle and tapering to nothing at the end. When the spindle is screwed home the thread on the spindle bottoms on the tapering thread of the nozzle and no oil can get to the tip. As the spindle is unscrewed the marked taper of the two threads cause them to separate and form in

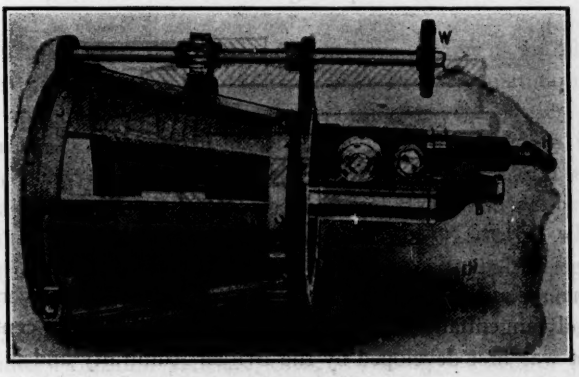



FIG. 3.

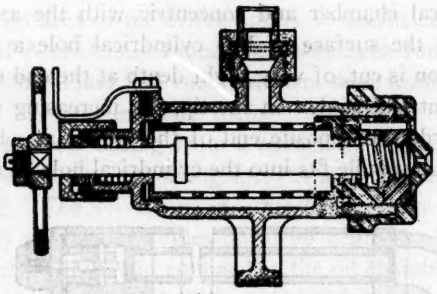


FIG. 4.

combination a spiral groove, the sectional area of which rapidly increases as the spindle continues to be unscrewed. The central chamber is formed by the combination of the end of the spindle and the burner tip. The output of the burner is controlled by the size of the outlet orifice and by the revolution of the spindle which regulates the area of the spiral oil passage.

### THE HOWDEN BURNER.

As will be seen, the Howden burner, Fig. 5, is almost a replica of the Schutte-Koerting burner in principle, except that the regulation of the spindle permits the output to be controlled. The helical passages are not affected, however.

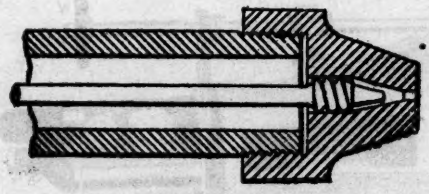


FIG. 5.—HOWDEN BURNER.

The second type of mechanical atomization by delivering the oil tangentially to a circular oil chamber with a central outlet was first developed by the British Admiralty and is the type that is most generally employed in mechanical-burner

construction. There are many claims as to priority of patents covering the mechanical burner, but not until the British Admiralty took hold of the matter was any combination of burners able to drive a boiler to full power and obtain good results. This they succeeded in doing.

The angle of the conical spray differs with different burners, depending on the size and the thickness of the walls of the outlet orifice. The thicker the wall the narrower the angle, and vice versa for a given size of outlet orifice. No burner, however, that I know of, produces a greater spray angle than 90 degrees. The angle of the spray is usually about 60 degrees to 70 degrees, and variation within these limits has little or nothing to do with the efficiency of atomization. As noted in a previous paragraph, atomization depends on the friction and other elements of the design. A mechanical burner that will atomize oil as finely as the air burner is desired.

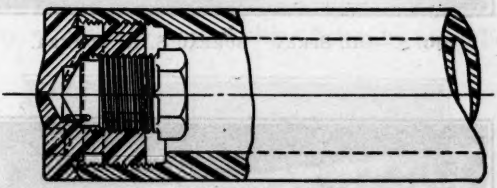


FIG. 6.—BUREAU S. E. BURNER.

The idea of some engineers is to have a valve as an inherent part of their burner, a high oil pressure being maintained up to and into the burner, and the output of the burner being throttled or regulated by this valve. This regulation is unnecessary and undesirable. The simpler the construction of the burner the more hard usage it will stand, and the better it will accomplish its purpose. It is very doubtful whether burners having a valve as an inherent part of their design can be regulated so that each will spray the same amount of oil. If, however, the burner has no valve, but is run wide open under a constant pressure, then, providing the central orifices or outlets are the same size, each burner must deliver the same

amount of oil per hour. This is of the greatest importance in properly regulating the air supply.

In attempting to construct a burner on the tangential principle it was found that four tangential passages to the circular chamber, each having the same area as the central outlet and inclined at an angle of 26 degrees to the horizontal, being supplied by four holes, each also of the same size as the central

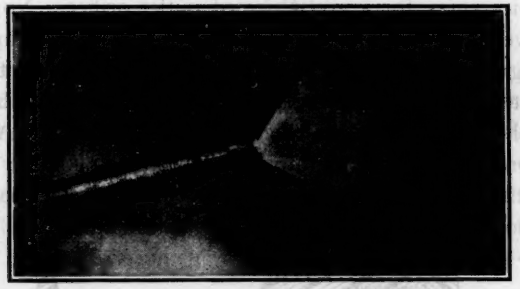


FIG. 7.—OIL SPRAY. BUREAU S. E. BURNER.

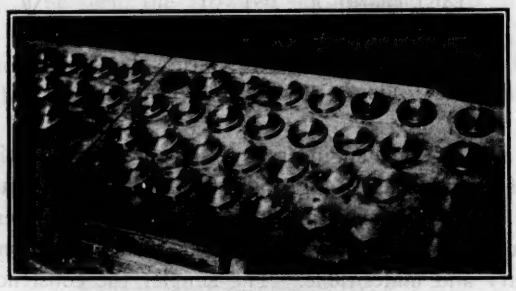


FIG. 8.—BURNER TIPS.

outlet, gave a burner with a very good spray, efficiency and capacity. It was also found that the higher the oil pressure within certain limits the finer the atomization. The burner, however, works very well under any pressure above 75 pounds. This burner is shown in Figs. 6, 7, 8, 9.

NORMAND BURNER.—Figs 10, 11.

This burner is installed on the *Paulding, Drayton, Trippe, Jouett, Jenkins,* and in general, all the Bath Iron Works boats.

Eight small holes, each the same size as the central outlet, deliver the oil tangentially to the central chamber formed by a

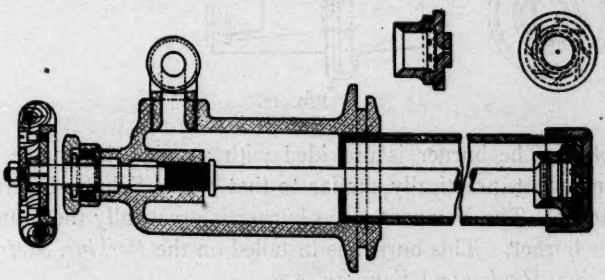


FIG. 10.

combination of the tip and the adjustable spindle. The oil chamber is recessed so that the movement of the spindle does not in any way close or affect these holes, its office being to

FIG. 11.

form a wall of the central chamber and to throttle or close the central outlet. The tip is made separately and is held in place on the burner pipe by a hexagonal nut.

FORE RIVER BURNER.—Fig. 12.

An adjustable spindle in this burner is arranged to throttle or close the central outlet and to vary the size of the central chamber to which the oil is delivered through two tangential

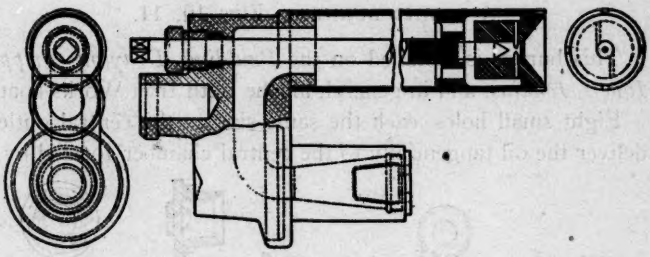


FIG. 12.

holes. The burner is provided with a quick detachable arrangement, practically similar to that of the Schutte-Koerting burner. The Newport News burner is practically the same as this burner. This burner is installed on the *Perkins, Sterrett, Walke, Henley* and *Fanning*.

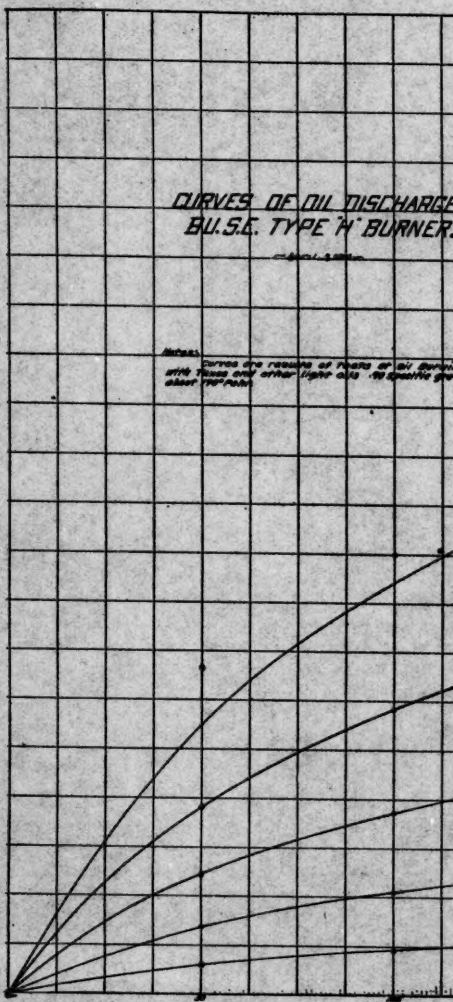
PEABODY BURNER.—Figs. 13, 14.

Oil is delivered under pressure to an annular chamber cut into the face of the nozzle upon which is screwed a tip having a small central chamber communicating with the central outlet.

Between the nozzle and the tip a thin disc is inserted and held in place. This disc has a hole in the center corresponding with the diameter of the central chamber, and small slots, four in number, extending tangentially from the edge of the central opening outward toward the periphery of the disc, long enough to overlap the annular channel of the nozzle and put it in communication with the central chamber.

FIG. 13.—PEABODY BURNER.

Fig. 15 gives the curves of capacities of the Bureau of Steam Engineering's burner with various size outlets used with oil of about 26 degrees B. gravity, heated to 170 degrees F. These curves can also be used for the Normand and Pea-


CURVES OF OIL DISCHARGE
B.U.S.E. TYPE 'H' BURNER.

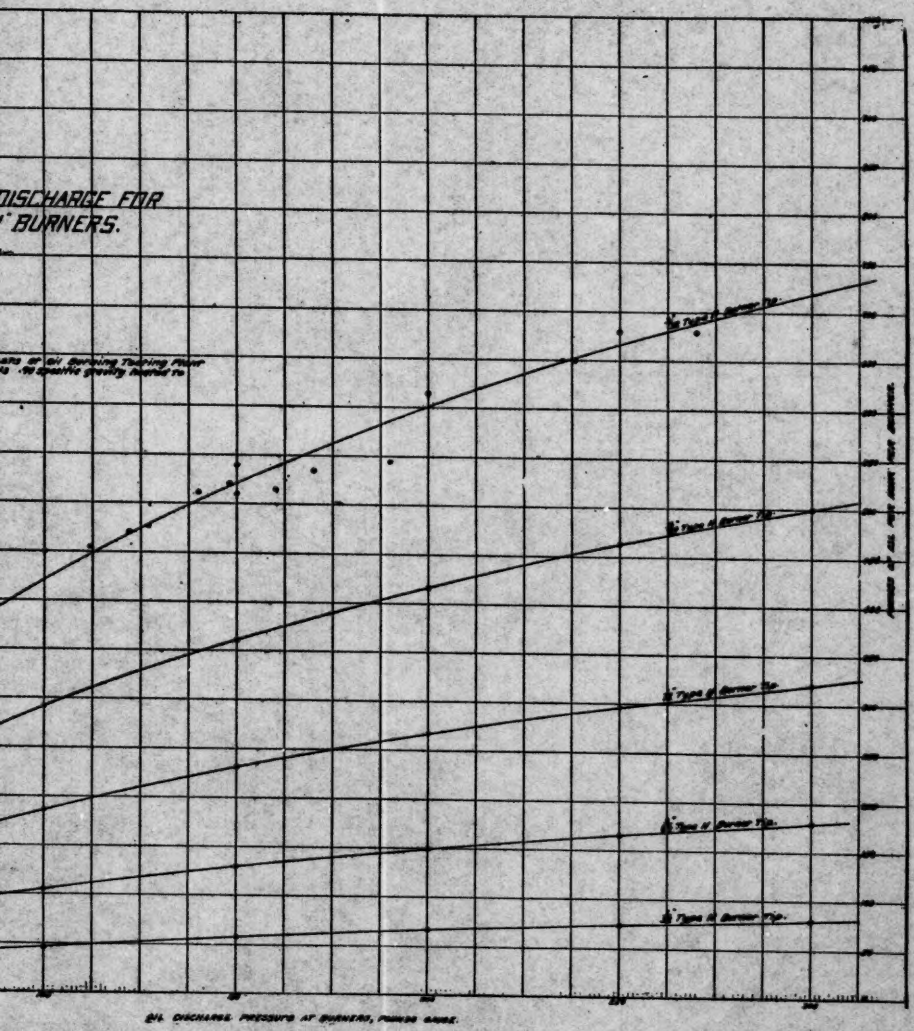



FIG. 15.

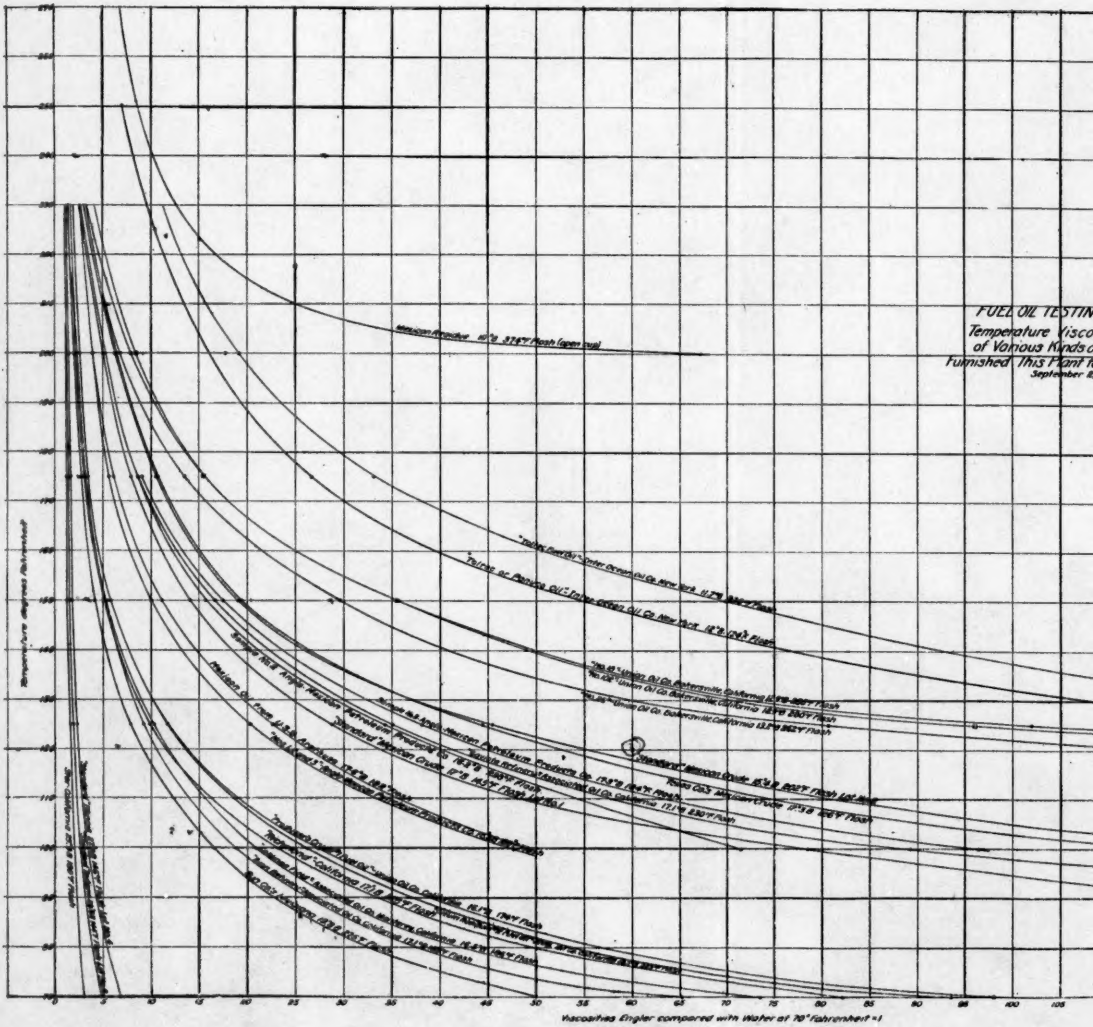
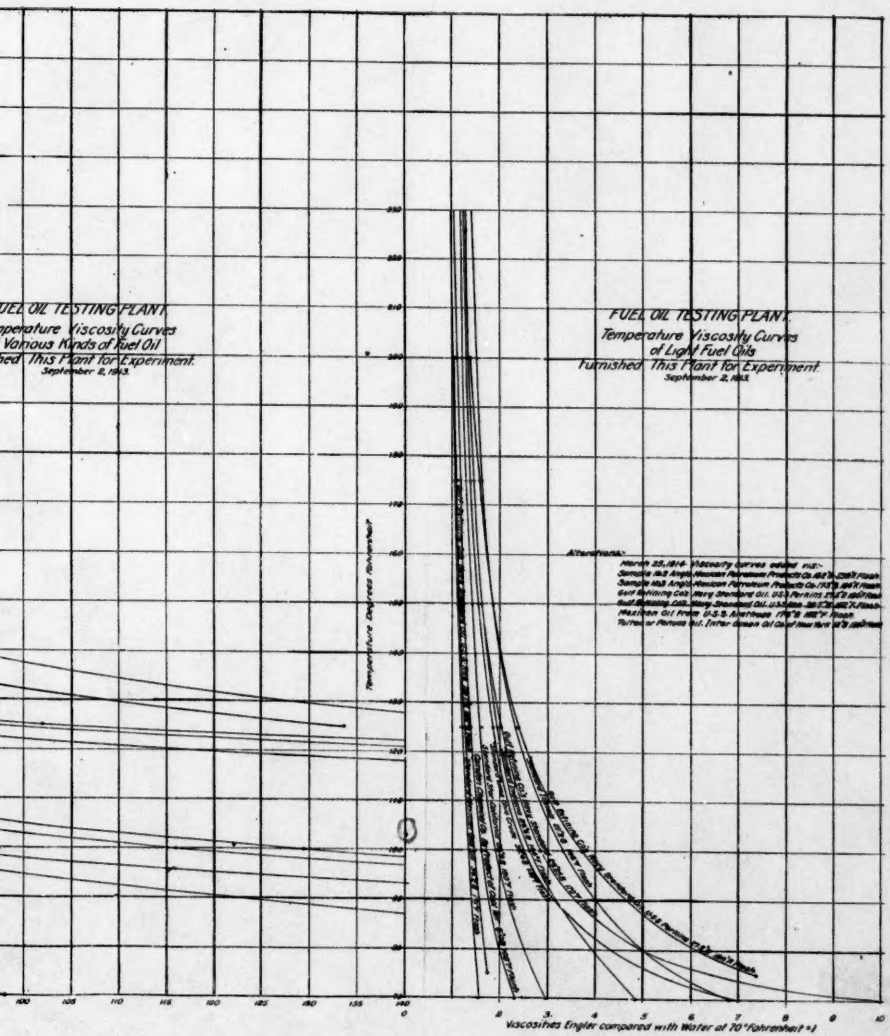

FUEL OIL TESTIN
Temperature Visco
of Various Kinds o
Furnished This Plant
September
Temperature degrees Fahrenheit
Viscosities Engler compared with Water at 70° Fahrenheit = 1

FIG. 16.


FUEL OIL TESTING PLANT.
Temperature Viscosity Curves
of Light Fuel Oils
Furnished This Plant for Experiment.
September 2, 1913.
Temperature Degrees Fahrenheit
Viscosities Engler compared with Water at 70° Fahrenheit = 1

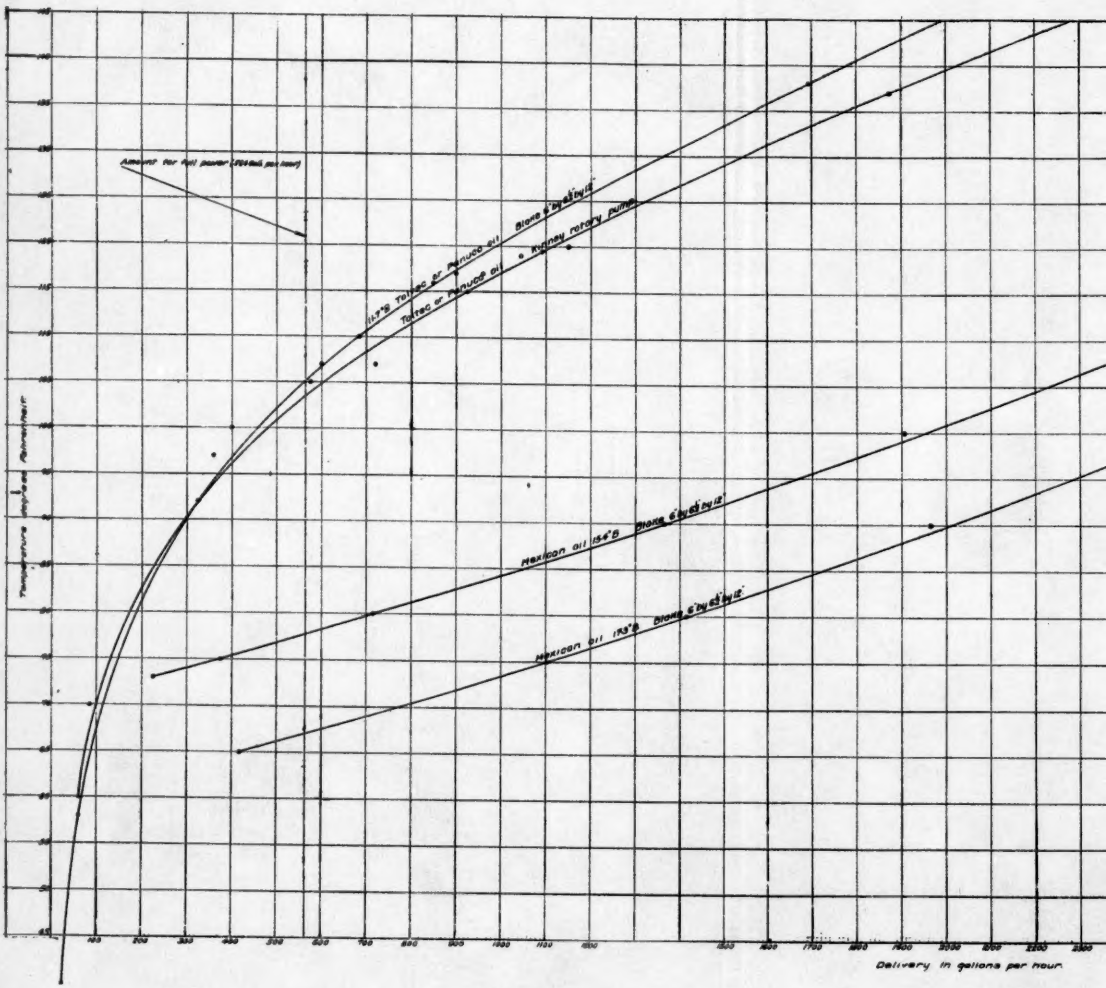
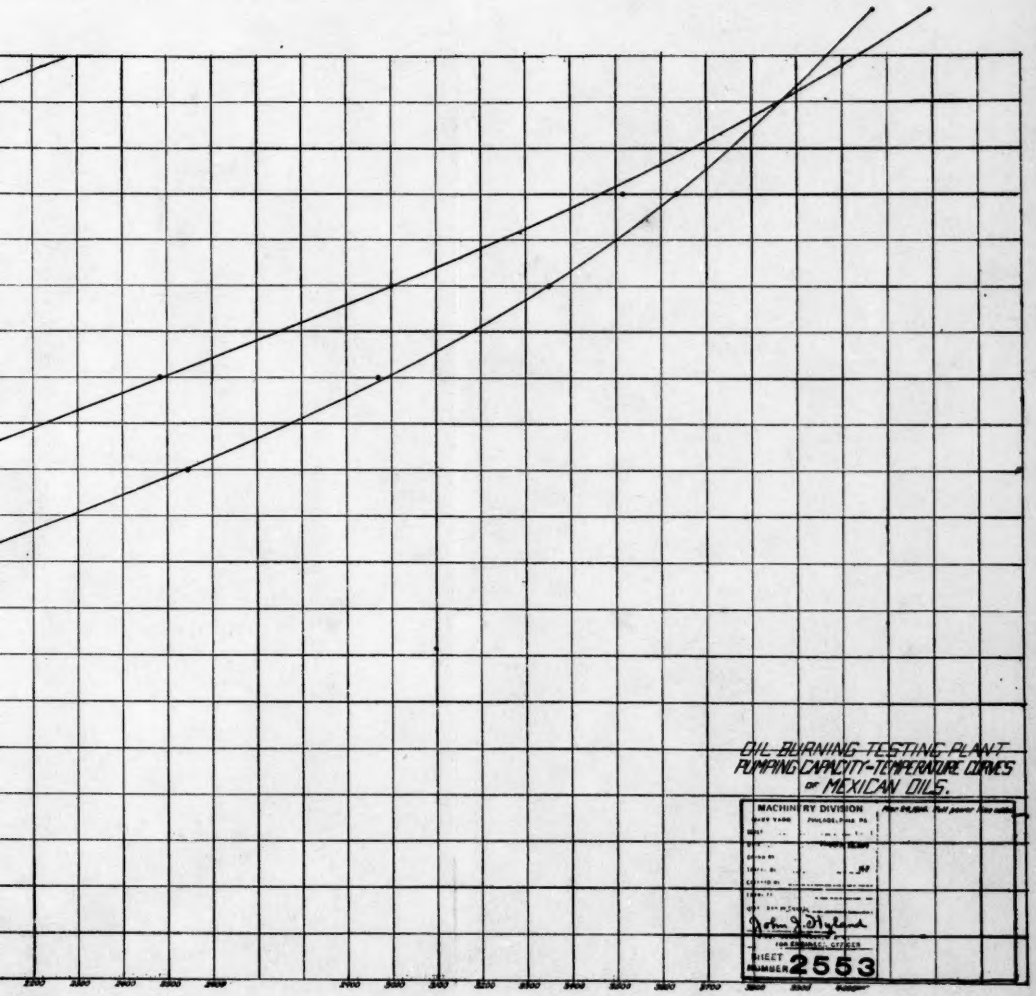

Amount for full power (Brake per hour)
113°B Tatao or Panuco oil Stone 6"by 6½"by 12"
Tatao or Panuco oil " Kinney rotary pumps
Mexican oil 154°B Stone 6"by 6½"by 12"
Mexican oil 193°B Stone 6"by 6½"by 12"
Delivery in gallons per hour.

FIG. 17.




OIL BURNING TESTING PLANT
PUMPING CAPACITY-TEMPERATURE CURVES
OF MEXICAN OILS.

MACHINERY DIVISION
SHOP YARD PHILADELPHIA PA
SHEET NUMBER 2553

Fig. 17.

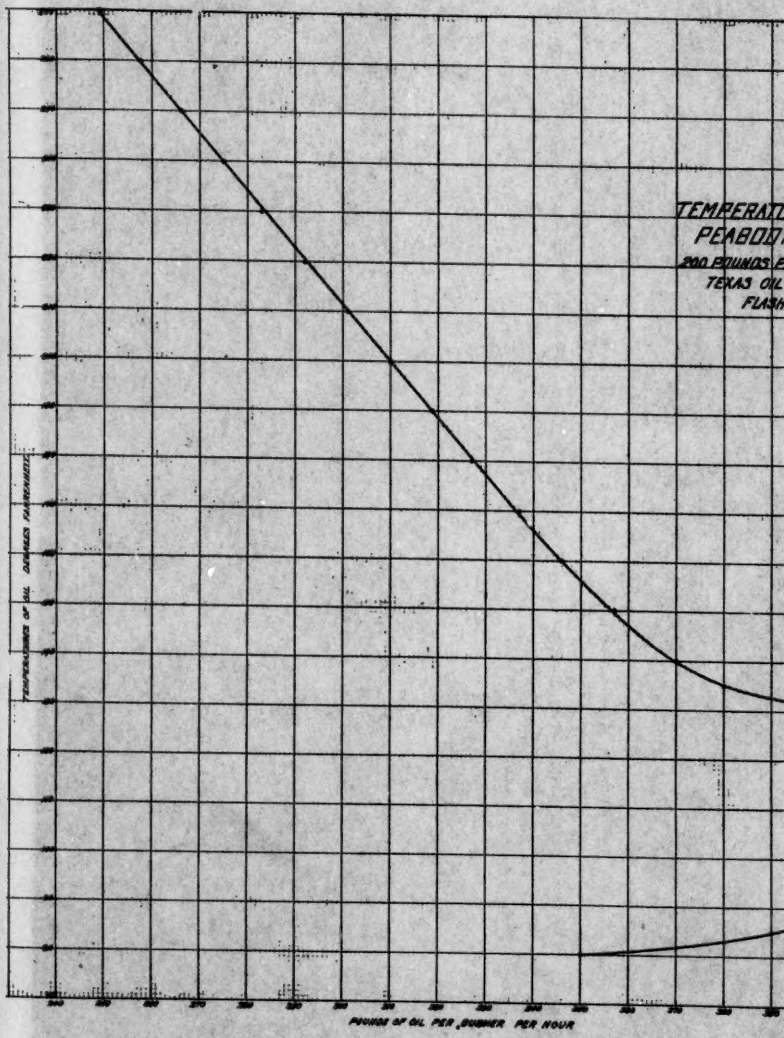
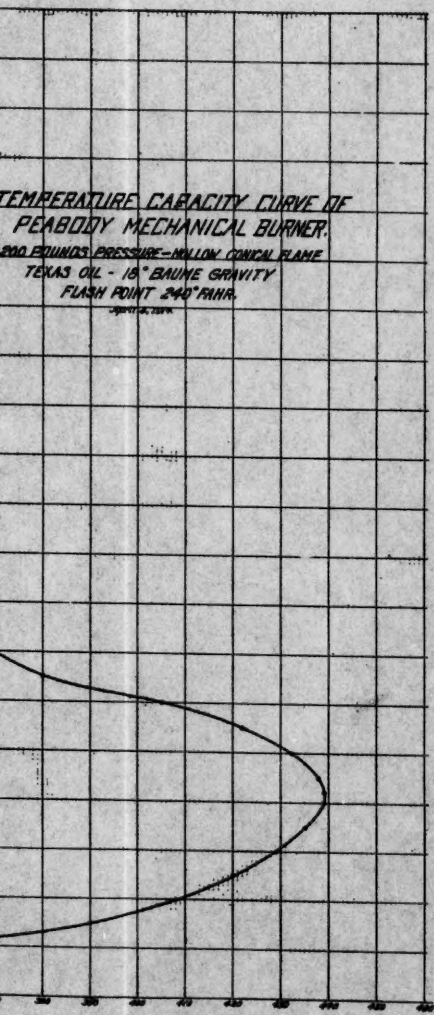

TEMPERATU
PEABOD
200 POUNDS P
TEXAS OIL
FLASH
TEMPERATURES OF OIL DEGREES FAHRENHEIT
POUNDS OF OIL PER BURNER PER HOUR

FIG. 18.


TEMPERATURE CAPACITY CURVE OF
PEABODY MECHANICAL BURNER.
200 POUNDS PRESSURE—HOLLOW CONICAL FLAME
TEXAS OIL - 18° BAUME GRAVITY
FLASH POINT 240° FAHR.

body burners. Naturally, new curves will have to be made for other oils of different gravity and temperature, but the above curves show the characteristic capacities of various burners used with light oils.

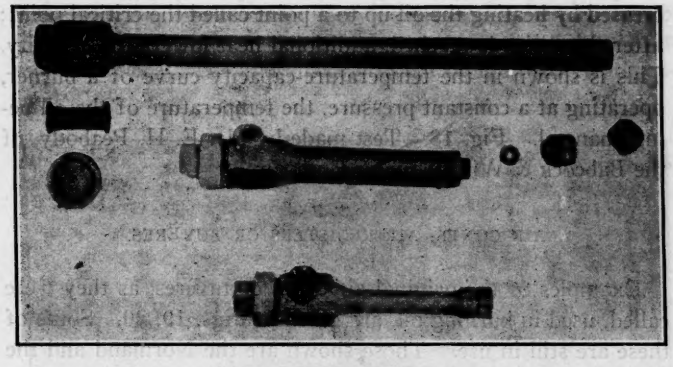


FIG. 14.—PEABODY BURNER.

As stated before the characteristics of the oils shown in the tables have very little to do with their proper burning, but we will now turn to the viscosities of oils, which is the main point to consider in the proper burning of an oil. Different oils from the various sections of the United States have been furnished the Fuel-Oil Testing Plant for experiment. The viscosities of these oils at various temperatures are shown in Fig. 16. This data practically covers the characteristics of oils from all over the world. Repeated tests have shown that in order to have good atomization and smokeless combustion the viscosity of the oil must be reduced by heating to 8 on the Engler scale. The use of additional heat to further lower the viscosity below 8 in no way improves the evaporation. The Pumping-Capacity Temperature Curves, Fig. 17, of three heavy Mexican oils is given, and it has been shown by repeated pumping tests that in order to be able to run full power, which is about 564 gallons per hour, the viscosity of the oil has to be reduced to 375, Engler scale, to produce this result.

An oil is heated for the purpose of reducing its viscosity, in order to be able to properly pump and atomize it. When the heating has accomplished this purpose the use of additional heat is of no value. Heating an oil also aids in disassociating any water that may be in it. The capacity of a burner is increased by heating the oil up to a point called the critical point; after this point is reached, additional heat lowers the capacity. This is shown in the temperature-capacity curve of a burner, operating at a constant pressure, the temperature of the oil being changed. Fig. 18. Test made by Mr. E. H. Peabody, of the Babcock & Wilcox Boiler Company.

### AIR CONES, AIR REGISTERS OR TUYÈRES.

Examples of the original air cones or tuyères, as they were called, used in burning oil, are given in Figs. 19, 20. Some of these are still in use. Those shown are the Normand and the Yarrow types. Part of the air entered at the front of the cone and part through the various slots or passages on the side. The air at the side has imparted to it a rotary motion,

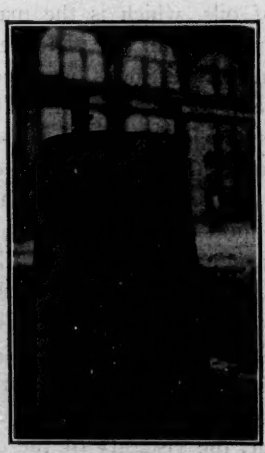


FIG. 19.—NORMAND TUYÈRE.

FIG. 20.—YARROW TUYÈRE.

that at the front has no rotary motion. With the Normand tuyère no regulation of the air except by means of draft doors was possible. In the Yarrow type an attempt is made to regulate the air entering at the front end of the cylinder by means of the flat circular plate shown, which was moved in or out and so varied the end opening. Except in the hands of very skilled operators, as a result of the imperfect air regulation vibration flare backs and shaking down of the brick work ensued.

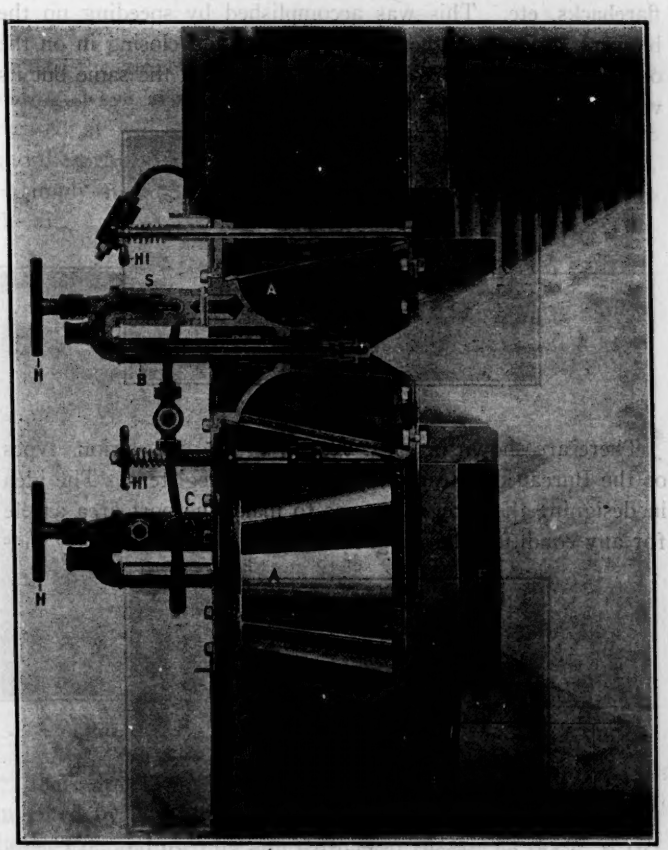



FIG. 21.—SCHUTTE-KOERTING AIR REGISTER.

The Schutte-Koerting register shown, Fig. 21, is one of the first attempts made to regulate the air supply by means of a sliding sleeve working on the outside surface of the frustum of a cone and operated by rack and pinion gear. The sleeve permits the air opening to be entirely closed, in case the burner is not in use, or to open it to fixed amounts. On the earlier oil-burning destroyers it was found that, in order to make this rig work properly, the air and the oil had to enter the furnace in certain relative velocities, in order to prevent vibration, flarebacks, etc. This was accomplished by speeding up the blowers, giving a higher static pressure and closing in on the openings so that the volume of air remained the same but its velocity was increased.

FIG. 22.

There are shown in Figs. 22, 23, 24, 25, 26, various types of the Bureau of Steam Engineering air registers. The idea in designing these registers was to make the port area ample for any condition that might be met, to impart a rotary mo-

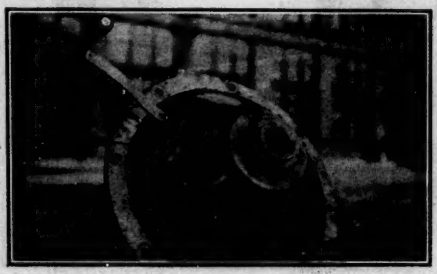


FIG. 23.

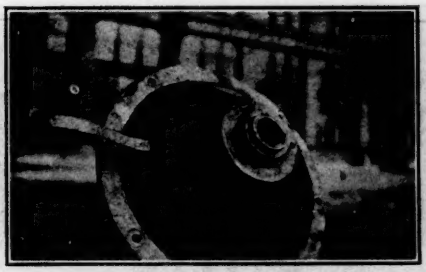


FIG. 24.

tion to the air and to adjust the port area by means of the sliding sleeve so as to permit the velocity of the air to be increased at will. The essential difference between this type and the Schutte-Koerting is that the latter is almost a cylinder, is simply an air distributer, and imparts no rotary motion to the air.

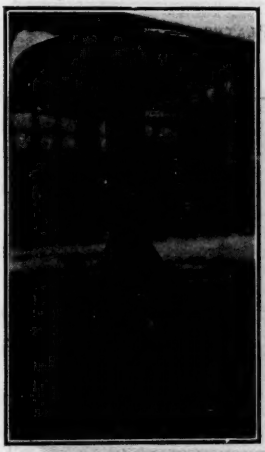


FIG. 25.

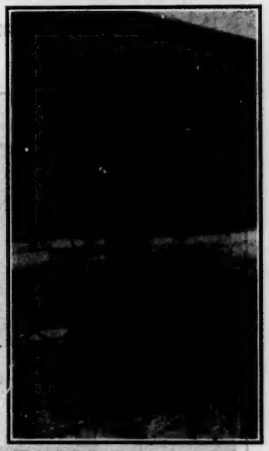


FIG. 26.

There are shown, Figs. 27, 28, 29, Peabody impeller plates, invented by Mr. E. H. Peabody, of the Babcock & Wilcox Boiler Co., and used by him with great success on the B. & W.

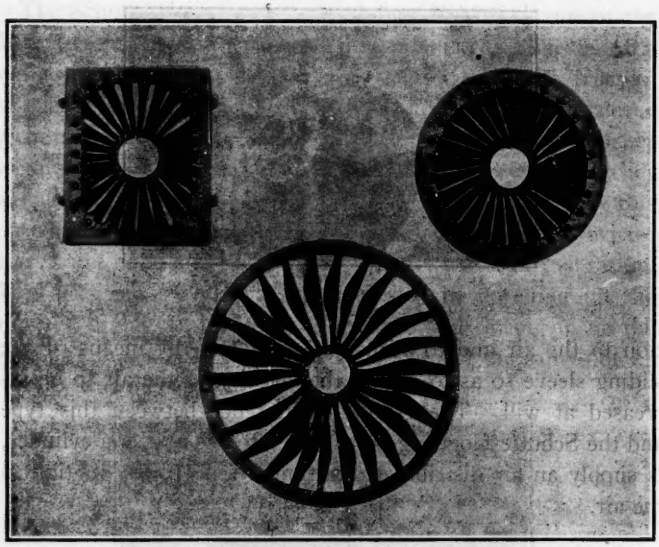


FIG. 27.—IMPELLER PLATES.

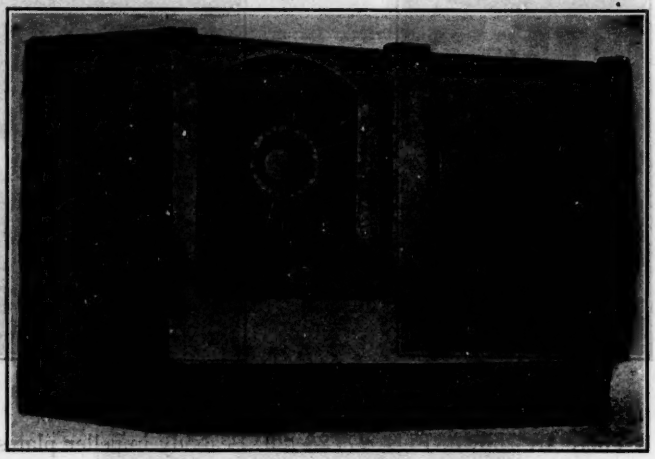


FIG. 28.—PEABODY IMPELLER PLATES. VIEW FROM FURNACE.

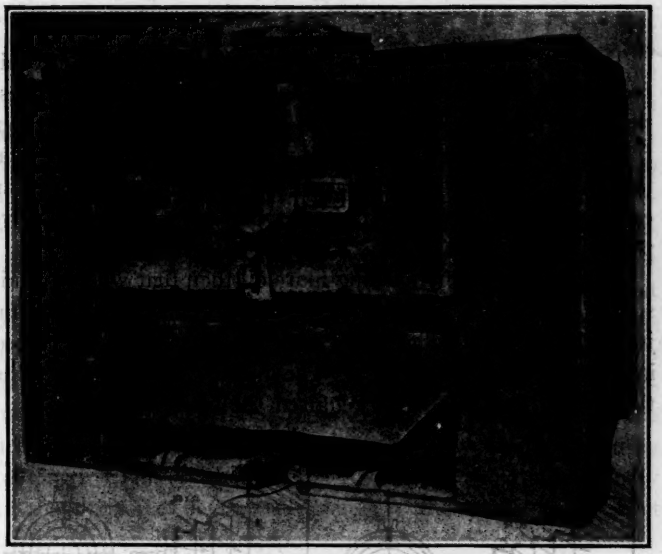


FIG. 29.—PEABODY BURNERS FOR COAL AND OIL. FRONT.

boilers, burning oil alone, or in combination with coal. The impeller is fixed, the vanes being set at an angle of 45 degrees. The air supplied to these impellers is regulated by draft doors on the oil-burning front. A rotary motion is imparted to the air. If the type of impeller is designed specially for the requirement of the boiler this impeller is as good as any on the market. There is one drawback: the plates which close off the impeller when not in use become red hot when closed and make it very hard to open them, once they are closed.

THORNYCROFT AIR DISTRIBUTER.—Fig. 30.

This, in general, is a system of concentric rings or short cylinders forming the frustum of a cone, the air for combustion being admitted through annular openings between the rings.

## AMOUNTS OF AIR FOR COMBUSTION.

To burn a pound of oil completely 15 pounds or 200 cubic feet of air are required. Attempting to burn oil with less than this amount of air results in incomplete combustion, forms a large amount of carbon monoxide, and causes vibration. In case a large amount of excess air is introduced an excess of oxygen results which will lower the evaporative power of the boiler.

Forcing a boiler by burning more oil than that which the air supply will consume smokelessly is only wasting fuel and injuring the boiler. These are facts that should be drummed into every one burning oil. The nearer we approach the theoretical quantity of air required the better the result.

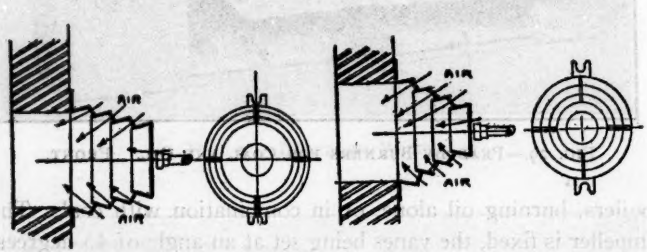



FIG. 30.

In burning oil the firerooms are generally closed up by means of air locks and airtight hatches. Turbine forced-draft blowers running as high as 1,800 r.p.m. supply the air from an outside louvre or inlet opening to the atmosphere, and deliver it into the boiler, where it mixes with the oil and forces the gases up through the boiler. The proper regulation of air is of prime importance. No air should enter the boiler at any place except through the air cones, registers or tuyères, where it will do useful work, and enter and mix with the oil spray in proper proportions. An airtight boiler casing is a necessity. I have found this so by continuous running on boilers for three years. All the joints, seams, etc., of the boiler casing should be made up with some sort of an asbestos

gasket or similar material. Poor evaporation is to a great extent due to leaky air casings. One method of examining for leaks is to turn the blower over slowly and examine all the seams with a lighted candle. If the flame is drawn into the inside of the casing, or put out, then the casing leaks. Another method is to sample the gases in the furnace and also at the base of the stack with an Orsat apparatus. If the results are similar, the casing is tight, if the stack sample indicates much lower $CO_2$ than the furnace, the casing leaks.

In order that data for design may be furnished the amount of air supplied to a boiler is measured. A Pitot tube is inserted in either the intake or discharge of the blower, where a uniform and steady flow of air can be obtained. This instrument measures the velocity of the air. Having this and the area of the inlet or outlet pipe, the quantity delivered can be obtained. This $= Q = V_1 \times A$, where $V_1 =$ mean velocity of the air in feet per second, and $A =$ the area in square feet of the blower discharge pipe.

This velocity $V_1 = \sqrt{2gH}$, where $H =$ velocity head in feet of air. One inch of water is equivalent to a pressure of 5.2 pounds per square foot.

$$\text{Then } H = 5.2 \times \frac{\text{Velocity head in inches of water}}{\text{Weight of one cubic foot of air}}.$$

Let

$h =$ velocity head in inches of water $=$ Dynamic head—static head;

$\rho =$ Density of air in pipe $=$ weight of one cubic foot of air;

$g =$ mean gravity $= 32.17$;

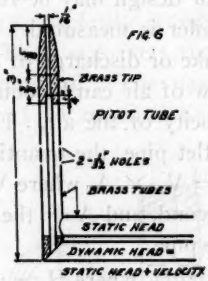
$V =$ velocity of air in feet per minute.

Then

$$H = 5.2 \times \frac{h}{\rho};$$

$$V = 60\sqrt{2gH} = 60\sqrt{\frac{64.34 \times 5.2 \times h}{\rho}} = 1{,}097\sqrt{\frac{h}{\rho}}.$$

FIG. 4



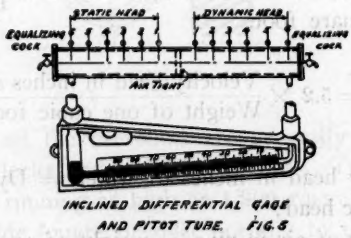




FIG. 31.

The velocity head is found by means of a differential gage attached, as shown in the sketch, Fig. 31.

## AIR CHAMBERS ON FUEL OIL LINES.

A necessary condition for properly burning oil is to have a uniform, steady pressure which will give a steady stream of oil into the furnace and prevent fluctuations in the discharg-

ing oil spray. Rotating plunger pumps have been built to accomplish this purpose. Two illustrations are shown of the above pumps, namely, the Kinney, Fig. 32, and the May-Nelson, Figs. 33, 34, 35. The principle of both is the same, namely, a disc set eccentrically on a shaft which has an eccentric movement imparted to it. They accomplish the purpose very well. Where these pumps are not provided, and none are in use in the Navy, air chambers on the dead ends of the line are employed. These air chambers are charged with air at 90 pounds' pressure per square inch, and the pump is started and the oil valve opened. If the air chamber is properly charged the fluctuation is practically eliminated. The oil, especially when heated, has a great affinity for oxygen, and it rapidly absorbs this element in the air, thus requiring the chamber to be recharged about twice a day.

## AIDS TO COMBUSTION.

The most useful guides to secure complete combustion of oil are (1) the use of the Orsat apparatus, (2) the amount of smoke produced, and (3) the color of the flame. In good combustion as high as 15 per cent. of $CO_2$ is obtained. About 2 to 2.5 per cent. oxygen, and about .2 of 1 per cent. to .4 of 1 per cent. of CO. One authority gives 2 per cent. to 8 per

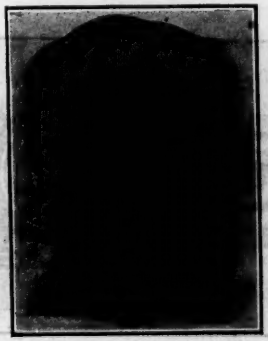


FIG. 32.

cent. oxygen as the requisite amount for good combustion. The total gives about 18 per cent., and the remainder, or 82 per cent., is called the nitrogen difference. If the total does not approximate 18 per cent. then something is wrong with the Orsat apparatus.

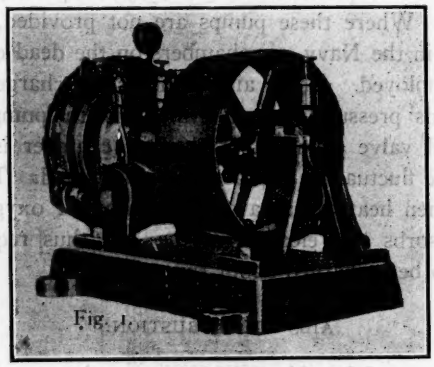


FIG. 33.

With a gas analysis as above, there should be very little or no smoke, about No. 1 on the Ringleman chart being the proper amount. Dense black smoke indicates incomplete combustion. No smoke indicates an excess of air. The Ringle-

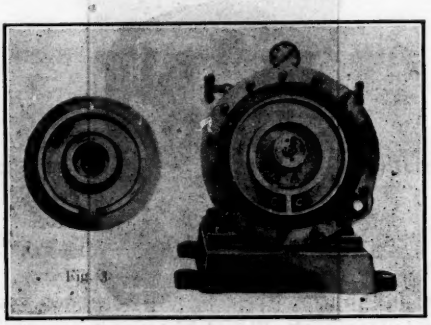


FIG. 34.

man chart is a scale or chart graduated from 0 to 5 and showing various differences from no smoke which, is " 0," to dense black smoke, which is " 5." By examining the flame a great deal can be learned. If the flame is pure white, and you can see the back walls through it, there is an excess of air; if the

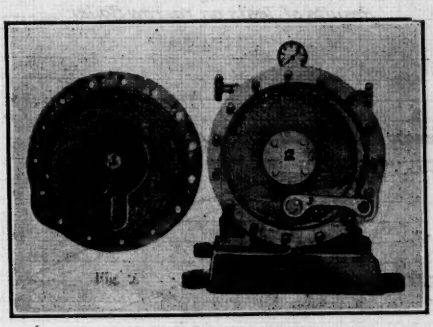


FIG. 35.

flame is yellowish-white, it is about right. If it is a dark yellowish red, there is insufficient air. An analysis of the gases by the use of the Orsat apparatus is, however, considered the best and surest indication of proper combustion.

## METHODS OF RUNNING UNDER AN OIL-FIRED BOILER.

Suppose no steam is on the plant, the boilers all being cold and no steam or air available. A hand pump is connected up to the discharge end of the burner line and the hand-pump suction taken from a five-gallon can filled with oil. The pump is started and the pressure brought up to and kept at about 200 pounds per square inch, the pressure being kept as steady as possible. For this purpose, the three-crank pump of the Schutte-Koerting Co. is a very good one. A torch made of a hooked rod with either waste or asbestos-ball wicking on the end is dipped in the oil. The addition of a little kerosene or gasoline to the torch improves matters. The torch is lighted; all the registers are opened wide to give as much air as pos-

sible to the boiler. Fortunately oil-burning boilers are not provided with dampers, which would be a menace in unskilled hands. The burner is opened and lighted and the pump kept going constantly. Unless the oil is a light oil whose viscosity

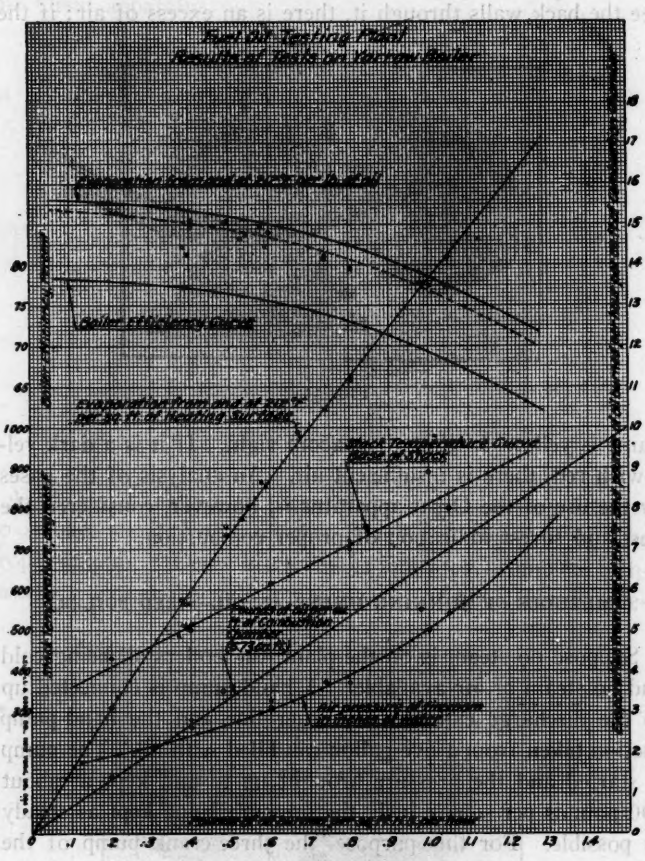


FIG. 36.

is 8 or below at 70 degrees F., the torch will have to be kept up to the burner constantly until the steam forms and the viscosity of the oil is reduced by heat to 8 or below, when the torch may be withdrawn. When steam pressure has reached

75 or 100 pounds, put steam on the oil pumps and heaters and cut out the hand pump. Put steam on the forced-draft blower and get ready to put on other burners; start blower slowly, open air registers wide and light off burner. As soon as it is

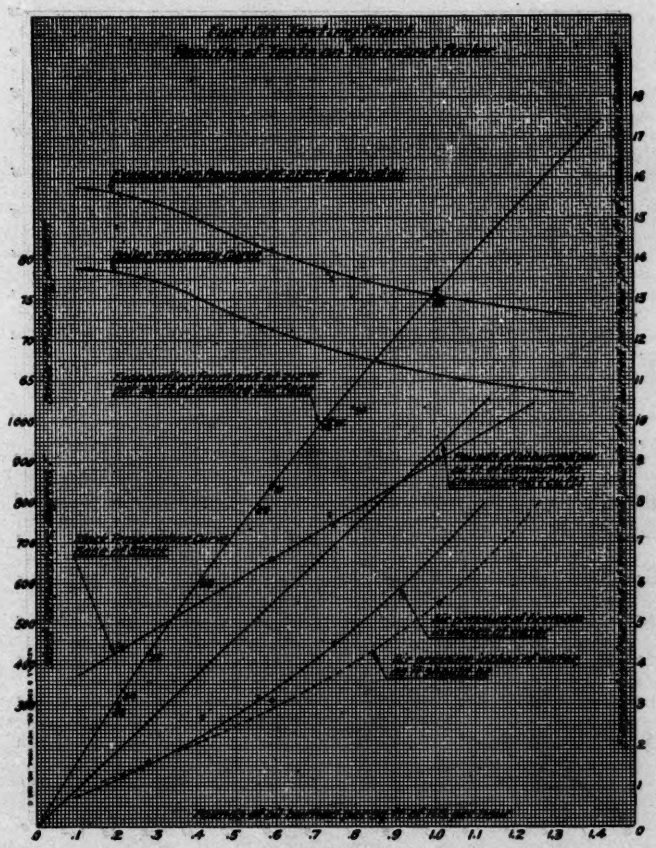


FIG. 37.

lighted, speed up the blower until you are sure the air pressure is sufficient to prevent vibration and flarebacks. Then set the vanes to the proper openings on the air registers. Always run with the air registers throttled down—it gives a better regu-

lation and a higher velocity to the air and ensures smoother running.

Until the boiler furnace is well heated an excess of air will be needed to prevent smoke. When the furnace becomes

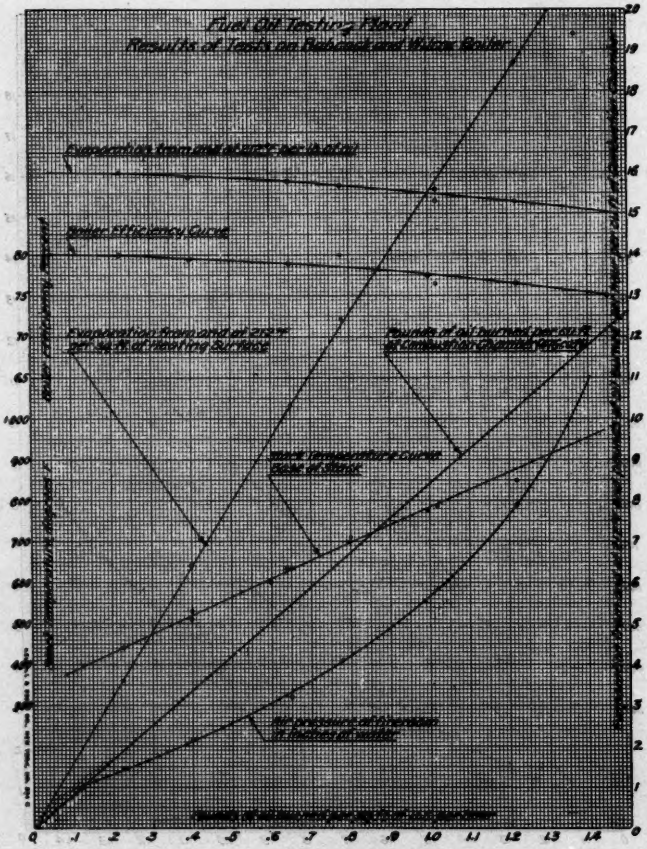

Fuel Oil Testing Plant
Results of Tests on Babcock and Wilcox Boiler
Evaporation from and at 212° per lb. of Oil
Boiler Efficiency Curve
Evaporation from and at 212° per lb. of Heating Surface
Pounds of Oil per Hour per sq. ft. of Combustion Chamber Capacity
Stack Temperature Curve
Draft at Stack
Air Pressure at Register in Inches of Water
Pounds of Oil per Hour per sq. ft. of Heating Surface

FIG. 38.

heated the air can be cut down to the proper amount. Cut in other burners as needed, first being sure the register is set and open to the proper amount. In case of vibration cut out a burner till the blower can be speeded up to give sufficient air.

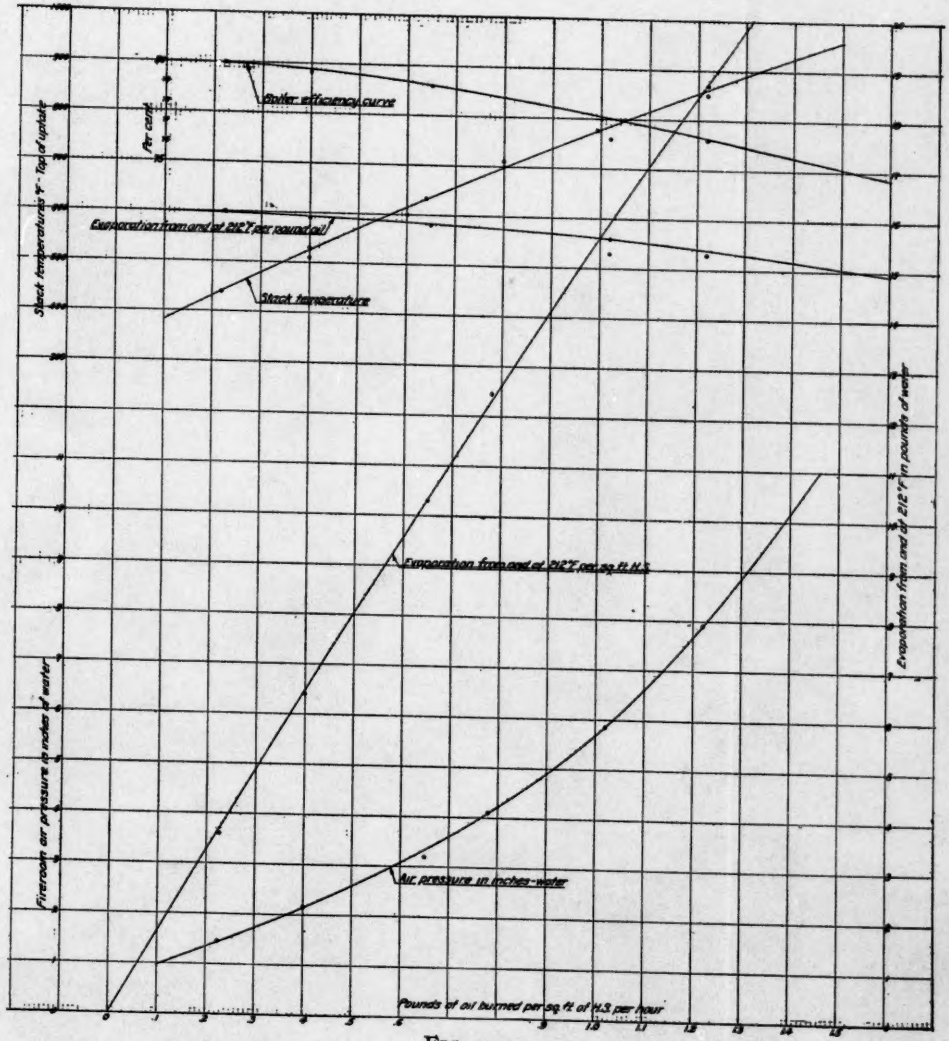
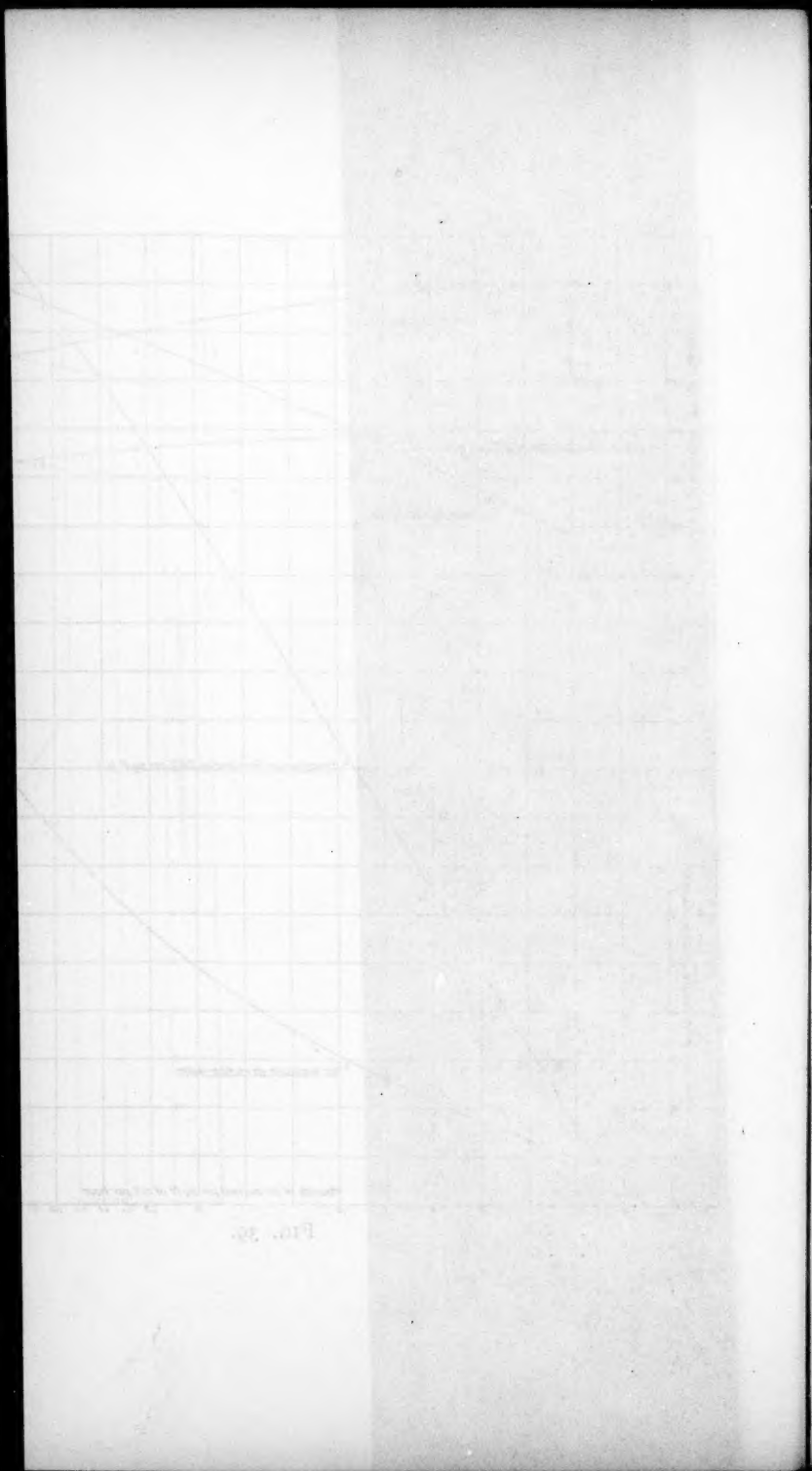

Stack temperatures °F - Top of upright
Per cent
Stoller efficiency curve
Evaporation from and at 212°F per pound oil
Stack temperature
Evaporation from and at 212°F per sq ft H.S.
Air pressure in inches-water
Pounds of oil burned per sq ft of H.S. per hour
Evaporation from and at 212°F in pounds of water
Furnace air pressure in inches of water

FIG. 39.



Run with very light smoke. Keep a half glass of water in the boiler; if the water level is too high the boiler will prime. It is a well known fact that as burners are added to a boiler, forcing it more, the water level rises, and vice versa. This should be provided for in operating the feed pumps. Never cut out all the burners at one time on a boiler operating at full speed. The water level will drop out of sight and give you many an anxious moment until you get it in sight again. In addition, operating with oils of 12 degrees B. to 17 degrees B. gravity, the relief valve of the oil pump shows a decided ten-

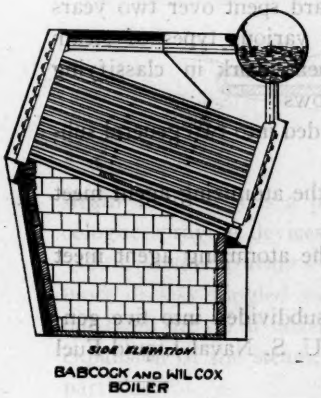



FIG. 40.

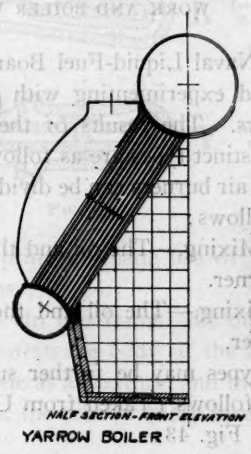



FIG. 41.

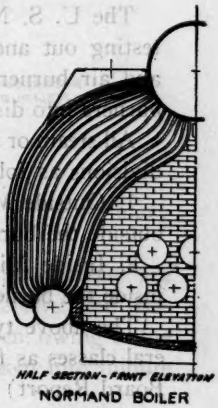



FIG. 42.

dency to stick. If you shut down a boiler suddenly with the oil pump running, there is danger of bursting the oil lines and causing a fire that might ruin the ship. Slow the pump down until the pressure is low enough to cut it out altogether; then close the burner valves.

There are shown herewith diagrammatic cuts, Figs. 36, 37, 38, 39, of the performances of the Yarrow, Normand, and Babcock & Wilcox oil-burning boilers at the Fuel-Oil Testing Plant. Recent tests at the Oil-Burning Testing Plant indicate that the curve of evaporation of the Yarrow and Nor-

mand boilers in pounds of water per pounds of oil can be raised about .4 pounds due to the improved burner made at the Fuel-Oil Testing Plant. It is very noticeable that the efficiency and general performance of the B. & W. boiler greatly exceeds the performance of the other two boilers, especially at the high rates. Figs. 40, 41, 42, show the general design of these boilers.

### STEAM OR AIR BURNERS, THEIR GENERAL CLASSIFICATION, AND THE VARIOUS TYPES USED BOTH FOR METALLURGICAL WORK AND BOILER WORK.

The U. S. Naval Liquid-Fuel Board spent over two years testing out and experimenting with various types of steam and air burners. The results of their work in classifying burners into distinct types are as follows:

All steam or air burners can be divided into two general subdivisions as follows:

1. Outside Mixing.—The oil and the atomizing agent meet outside the burner.

2. Inside Mixing.—The oil and the atomizing agent meet inside the burner.

The above types may be further subdivided into five general classes as follows (Taken from U. S. Naval Liquid-Fuel Board Report) Fig. 43:

1. Drooling Burner.
2. Atomizer Burner.
3. Chamber Burner.
4. Injector Burner.
5. Projector Burner.

1. *Drooling Burner.*—The oil supply simply oozes or drools out, at the orifice over and on to the steam or air jet. The action is as follows: As the steam or air issues forth it expands within the layer or film of oil which is being carried into the furnace.

2. *Atomizer Burner.*—In this burner the oil is brought through an orifice from which it is swept off by a brush of

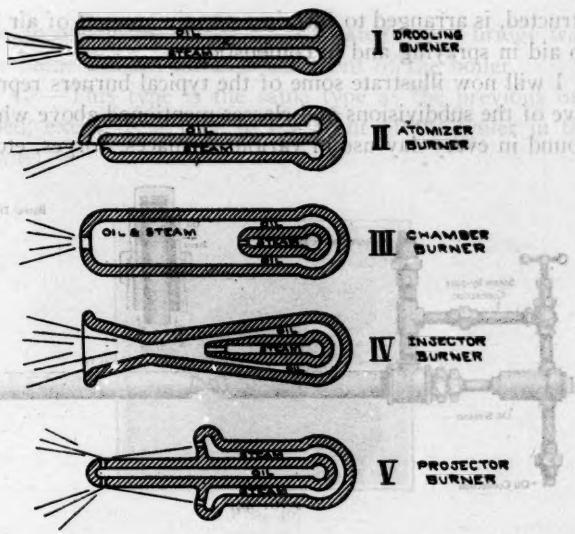



FIG. 43.

steam or air. It is a principle made use of in the ordinary cologne spraying devices.

3. *Chamber Burner.*—In this burner the oil and steam are more or less mingled within the body of the burner and pass out from the tip or nozzle as a mixture, and then, owing to the expansion of the steam, the oil is rapidly broken into minute particles.

4. *Injector Burner.*—Burners of this type are analogous to the injector used for boiler feeding and other purposes. Here the steam and oil rising, each through its own passage, mingle within cone-shaped passages and, as a mixture, passes through a contracted nozzle and then outward through a reversed flaring cone.

5. *Projector Burner.*—In burners of this type the oil is pumped to the oil orifice and from there is caught by a passing gust of steam and blown off. This might be regarded as a sub-classification of No. 2, the atomizer burner, except for the fact that the brush of steam is located some distance from the oil orifice, and this sweeping brush of steam, as usually con-

structed, is arranged to entrain a certain amount of air further to aid in spraying and in combustion.

I will now illustrate some of the typical burners representative of the subdivisions and classes mentioned above which are found in every-day use in various furnaces, forges, etc.

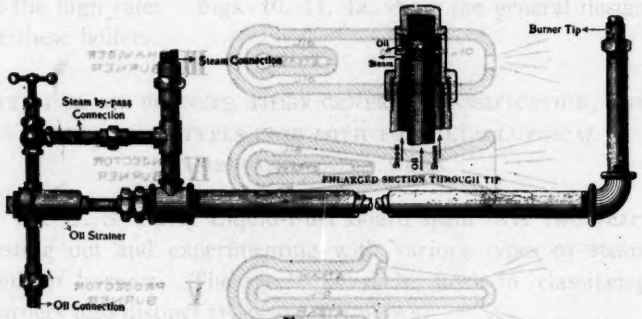



FIG. 44.

*Peabody Burner,* Fig. 44.—Outside mixing, drooling, giving a fan-shaped flame and spraying from a sharp edge. The cut shows the construction, including oil strainers, steam by-pass and removable tip. This tip contains two very narrow slots separated by a diaphragm, the upper slot for oil and the lower one for steam. The oil falls at right angles upon the steam jet which atomizes it and the oil is burned in a fan-shaped flame. The mixing is done entirely outside of the burner. This burner is used in connection with the Peabody

FIG. 45.

furnace, in which the burner tip is located at the bridge wall and the flame thrown towards the front of the boiler.

Fig. 45.—This type is the same type as the previous one described, except it is used at the front of the boiler in the usual way.

FIG. 46.

*W. N. Best High Pressure Oil Burner,* Figs. 46, 47.—Outside mixing, atomizing burner. Long slot spraying from a

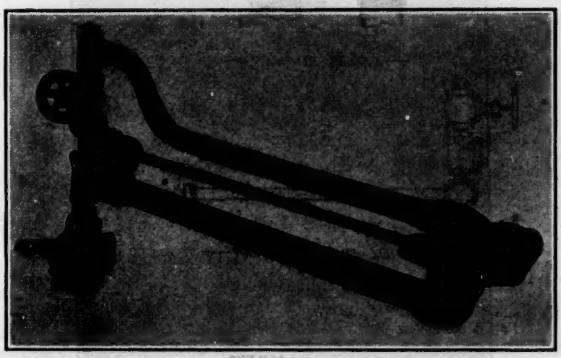


FIG. 47.

sharp edge. The air or steam meets the oil at right angles, atomizing it externally. Either a long narrow flame or a fan-shaped flame can be provided as required.

*Gem Oil Burner,* Fig. 48.—Outside mixing, drooling, rose-shaped orifice. Spraying aided by centrifugal action of the

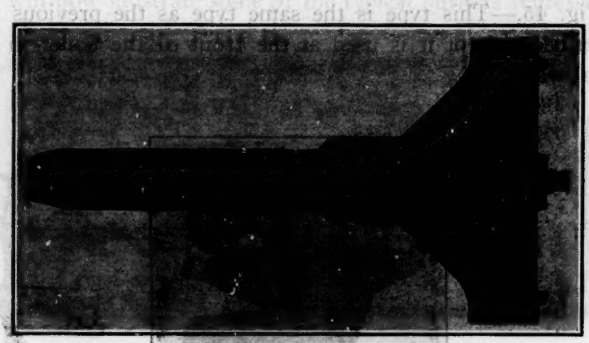


FIG. 48.

steam or air on internal helical screw thread. Used on brick kilns, etc.

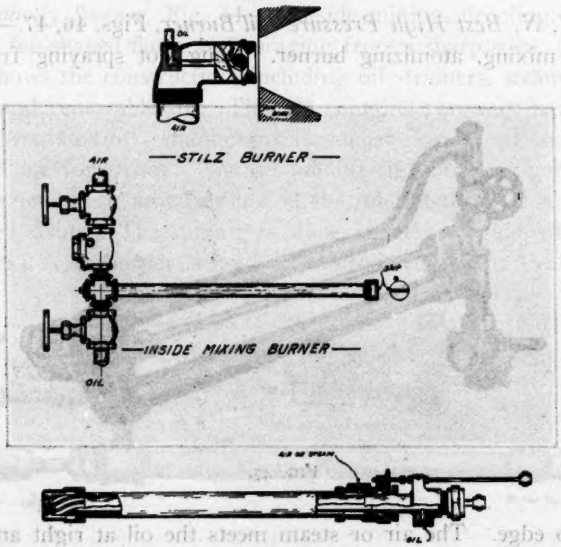



FIG. 49.

*Stilz Oil Burner,* Fig. 49.—A general adaptation of the Schutte-Koerting mechanical burner and the Gem oil burner, the internal helix to atomize and break up the oil, the outside one for the air or steam. Used on brick kilns, etc.

FIG. 50.

*Gilbert and Barker Oil Burner,* Fig. 50.—Outside mixing, drooling, rose-shaped orifice. The spraying aided by slight centrifugal action from the internal helix. Adapted for metallurgical operations, brick kilns, etc.

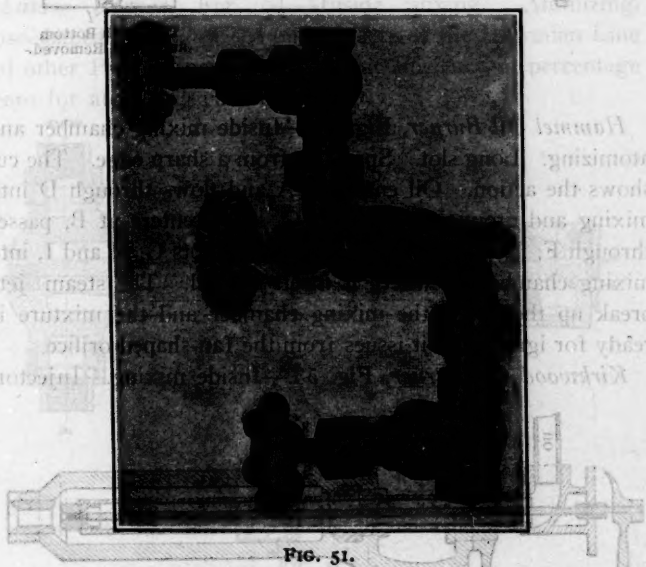


FIG. 51.

*Rockwell High-Pressure Oil Burner,* Fig. 51.—Outside mixing, atomizing. Steam or air and oil nozzles removable for cleaning. Steam or air nozzle outside of oil nozzle.

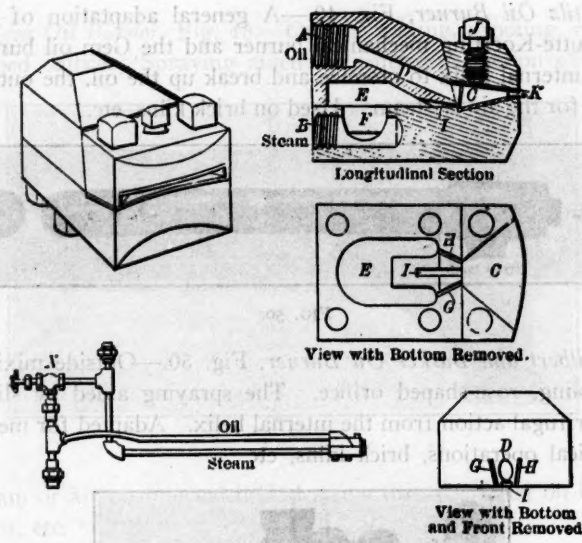



FIG. 52.

*Hammel Oil Burner,* Fig. 52.—Inside mixing chamber and atomizing. Long slot. Spraying from a sharp edge. The cut shows the action. Oil enters at A and flows through D into mixing and atomizing chamber C, steam enters at B, passes through F, E, and then through 3 small slots G, H and I, into mixing chamber C, where it meets the oil. The steam jets break up the oil in the mixing chamber and the mixture is ready for ignition as it issues from the fan-shaped orifice.

*Kirkwood Oil Burner,* Fig. 53.—Inside mixing. Injector.

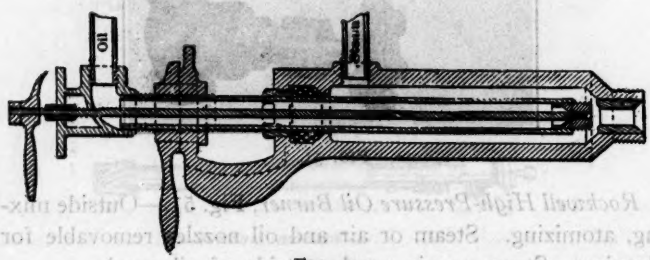



FIG. 53.

Rose-shaped orifice spraying from a sharp edge. Both oil and steam valves seat at the tip of the burner.

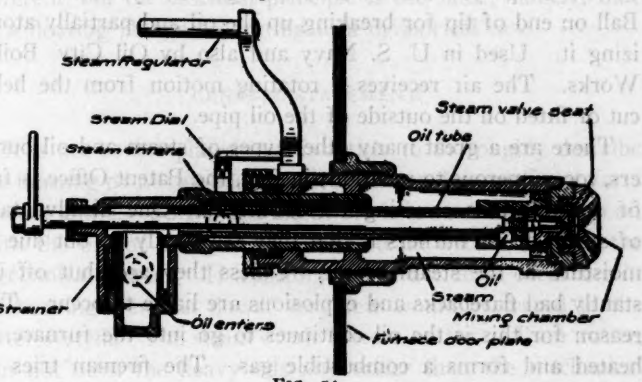



FIG. 54.

*Lassoe Burner,* Fig. 54.—Inside mixing. Atomizing. Rose-shaped orifice. Used on steamers of the Hawaiian Line and other Pacific Coast boats. Great efficiency in percentage steam for atomizing claimed for it.

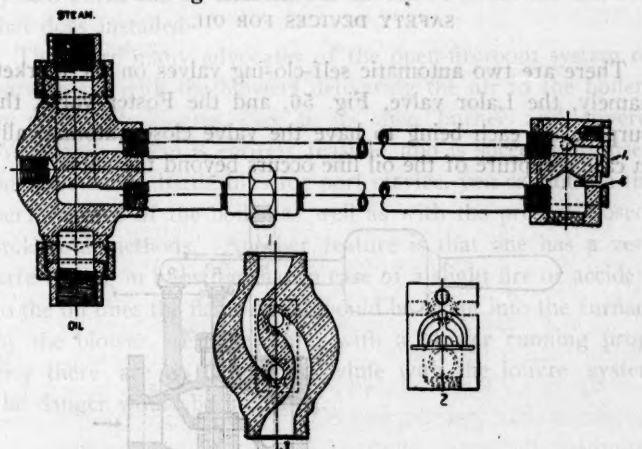



FIG. 55.

*Ingram Burner,* Fig. 55.—Outside mixing. Atomizing. Fan-shaped flame. Installed on various destroyers for port use.

*Fitton Burner,* Fig. 49.—Outside mixer. Atomizer. Rose or conical spray. Oil and air each regulated by its own valve. Ball on end of tip for breaking up the oil and partially atomizing it. Used in U. S. Navy and also by Oil City Boiler Works. The air receives a rotating motion from the helix cut or fitted on the outside of the oil pipe.

There are a great many other types of steam and oil burners, too numerous to mention; in fact, the Patent Office is full of specifications covering these burners. One disadvantage of steam or air burners is that they frequently go out due to moisture in the steam or air. Unless they are shut off instantly bad flarebacks and explosions are liable to occur. The reason for this is the oil continues to go into the furnace, is heated and forms a combustible gas. The fireman tries to light the burner from the back wall, the furnace not being blown out sufficiently and attempting to relight the burners, in many cases sufficient air is introduced to form an explosive mixture and an explosion or flareback results.

### SAFETY DEVICES FOR OIL.

There are two automatic self-closing valves on the market, namely, the Lalor valve, Fig. 56, and the Foster valve, the purpose of each being to have the valve close automatically in case a rupture of the oil line occurs beyond the valve.

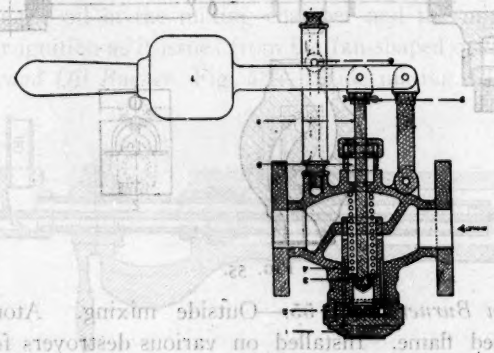


FIG. 56.

Some of the mechanical features of these two valves are different, but the essential principle is the same, namely, that of a floating plunger operating in a cylindrical case.

## GENERAL STATEMENT.

It is not contemplated in this article to thoroughly describe the burning of oils in boilers. However, a few pertinent facts are not amiss. So far as I am aware it is no easier to burn oil with the use of compressed air or steam than it is with the mechanical system. If a history of the accidents occurring in oil burning were set down, very few would be traced to the latter system. The steam or air system, so far as I am aware, can not burn the heavy oils any easier than the mechanical system. I think it will be found that the viscosity of the oil has to be about the same with either system. Practically any apparatus can burn oil, but the main point is, "does it burn it efficiently?" On this rests the whole subject, for unless the system burns the oil efficiently it should be discarded and one that does installed.

There are many advocates of the open-fireroom system of burning oil with the blowers delivering the air to the boilers by means of louvres, each to its own burner and tuyère. While this system is entirely feasible, and is successfully used on some of the battleships for port service, you cannot see the performance of the boiler as well as with the present closed-stokehold methods. Another feature is that one has a very safe feeling in knowing that in case of a slight fire or accident to the oil lines the flame or oil would be swept into the furnace by the blower. Furthermore, with a blower running properly there are no flarebacks; while with the louvre system the danger would be increased.

# METHODS OF SECURING ECONOMY IN STEAM CONSUMPTION.—SUPERHEATING AND IMPROVED CONDENSING APPARATUS.

By Lieut. Commander H. C. Dinger, U. S. N., Member.

## SUPERHEATING.

The object of the superheater is to give the steam after being formed in the steam drum an additional amount of heat so that the steam will work more efficiently in the engine. A pound of steam as made in a boiler contains a certain quantity of heat, measured in heat units. At 250 pounds pressure with dry steam the heat units are 1,204, and the temperature is 401 degrees F., at 200 pounds; heat units 1,198, temperature 381 degrees F. Now when this steam works in the engine it changes some of the heat units into work and when leaving the engine, by the exhaust pipe, it still has a considerable amount of the heat left. The exact amount depending on the vacuum and the design of engine, the better the vacuum the less heat there will be left and the more the engine will take up and convert into work. This heat that is left in the steam is taken away by the circulating water of the condenser and is thus a loss, most of which is now unavoidable. Now if we pass the steam after leaving the steam drum through a superheater we will give it more heat per unit weight than it had before. Thus if we take the steam at 250 pounds with a temperature of 401 degrees F. and heat it to a temperature of 501 degrees F., that is giving it 100 degrees superheat, the total amount of heat carried to the engine by a pound of steam will be about 1,269 heat units instead of 1,204 heat units with saturated steam. So the steam in the superheated condition will be able to do considerable more

work and a much larger part of the total heat from each pound of steam will be utilized by the engine. (The heat rejected to the condenser being approximately the same.) Hence by superheating a considerable increase in economy is obtained, since a larger percentage of the heat imparted to the steam is utilized to produce mechanical work.

This may be illustrated by an approximate example. Taking steam at 250 pounds absolute pressure, the total heat is equal to 1,204 B.t.u. If 840 B.t.u. are rejected at the condenser the available heat is $1,204 - 840 = 364$. If steam is superheated 100 degrees F., the total heat will be approximately 1,269 B.t.u., and if, as before, 840 B.t.u. are rejected to the condenser the available heat is $1,269 - 840 = 429$. The increase in available heat in proportion to that supplied is in this case about 12 per cent. If there were no heat losses in the engine this would represent the expected gain from the superheat. But as there are losses, the actual gain is somewhat greater, since some of the losses, such as initial condensation, are relatively diminished by the steam being superheated. On the other hand, radiation is probably increased owing to the use of higher temperatures, so that the real practical gain in reduction of steam consumption per I.H.P. is about 12 per cent. for 100 degrees superheat and an initial pressure of 250 pounds.

The theoretical gain in efficiency based on the efficiency rules for a perfect heat engine would be as follows:

$$\text{Efficiency} = \frac{T_1 - T_2}{T_0},$$

in which $T_1$ is initial temperature,

$T_2$ is final temperature,

$T_0$ is absolute temperature.

In this case $T_1$ equals 401 degrees for saturated steam at 250 pounds; 501 degrees for superheated steam at 250 pounds.

$T_2 = 102°$ temperature corresponding to a pressure of 1 lb. abs.

$T_0 = 401$ degrees + 461 degrees for saturated steam.

$T_0 = 501$ degrees + 461 for superheated steam.

For saturated steam; Efficiency $= \frac{401 - 102}{862} = .347.$

For superheated steam; Efficiency $= \frac{501 - 102}{962} = .415.$

The gain in efficiency is here shown to be about 19 per cent. This represents the theoretical gain in economy secured on a perfect heat engine, by the use of 100 degrees superheat, the engine using 250 pounds absolute initial pressure and exhausting at 1 pound absolute. According to this, superheating 100 degrees would give the same increase in economy that would be secured by using an initial pressure of 650 pounds.

Another method of considering the gain would be as follows: A good reciprocating engine using saturated steam as above, will require about 14 pounds of steam per I.H.P. This result represents the utilization of 183 heat units actually converted into work from every pound of steam used. As the available heat is 364 B.t.u. the engine efficiency is slightly over 50 per cent. and about one-half of the available heat is in some way lost.

If we assume that the amount of heat lost in the engine and condenser is the same in each case, the gain to be expected from the use of superheat may be calculated as follows:

*Saturated*, 364 heat available — 181 heat lost in engine = 183 heat used.

*Superheated*, 429 heat available — 181 heat lost in engine = 248 heat used.

The actual heat furnished by the boiler is for, saturated steam, 1,204 — 200 (heat of feed) = 1,004; for superheated steam, 1,269 — 200 = 1,069.

Efficiency, saturated $= \frac{183}{1{,}004} = .1824.$

Efficiency, superheated $= \frac{248}{1{,}069} = .234.$

This indicates a theoretical possible gain of about 27 per cent. due to the use of 100 degrees superheat.

The assumption that the losses in the engine are the same is, however, in error, since radiation losses certainly are increased by the use of higher temperature. The assumption that the heat rejected to the condenser is the same in each case is also, no doubt, somewhat in error, since it is probable that with the steam starting out in the superheated condition it will enter the condenser in a somewhat drier state than when starting out in the saturated condition. If these errors are considered as having the effect of several per cent. it will be seen that this analysis shows a result that approximates to the theoretical gain in efficiency shown by considering the cycle of the perfect heat engine.

In practice, by actual test, 100 degrees superheat under the conditions considered, has secured as much as 12 per cent. increase in fuel economy. If we consider the efficiency on the basis of the available heat in each case, results are as follows:

$$\text{Efficiency, saturated} = \frac{183}{364} = .505;$$

$$\text{Efficiency, superheated} = \frac{248}{429} = .578.$$

Gain in efficiency 14.4 per cent.

This approximates closely to the actual gain found in practice.

### GENERAL COMMENT AND DIFFICULTIES ENCOUNTERED.

There is now no question concerning the enormous gain in economy that may be secured by the use of superheat on marine boilers; nor is there any difficulty attending the practical operation of the superheater. This fact has been demonstrated by years of satisfactory service.

The fitting of superheaters does not add to the boiler weights since the increased economy secured permits of there being less total heating surface for the same power; so that the weight required for boilers is actually less. There is now no essential increase in space required, since the superheating tubes are installed in spaces not otherwise used.

## DIFFICULTIES THAT MAY BE ENCOUNTERED.

The chief difficulty that may be encountered with superheat is that owing to the higher temperature used the piping and fittings must be of material that will stand this temperature. Good bronze or steel castings and steel piping will stand a temperature of 500 degrees F. without question. For the large piping, steel should be used and steel or Monel metal valves and fittings. For small piping good, heavy copper or composition pipe will safely withstand the 500 degrees. Good, heavy composition castings can be used for the smaller fittings without question. For valve seats it is advisable to employ Monel metal and bronze valve discs. Steel valve discs will erode much more rapidly than bronze with either saturated or superheated steam.

Another place where difficulty may be feared is in the H.P. valves and cylinders. This has also been safely overcome by employing strong, hard cast-iron piston and valve packing, well designed, and admitting some lubrication to the wearing surfaces. The lubrication of the surfaces may be secured by use of graphite feeder and the coating of the wearing surfaces with graphite grease or vaseline when opportunity offers for opening up between runs.

Difficulties due to heat, of course, become more serious as higher temperatures are used, but with good, strong and sound material these difficulties are more imaginary than real, and are no greater with superheat than they are with high-pressure saturated steam.

In European shore practice superheating to temperatures of 700 degrees is successfully carried out. So that the objection to use of superheating in marine practice on the ground of difficulty due to fatigue of material from high temperature cannot be said to be well founded. The disinclination to utilize the advantages of superheat in marine practice is a conservatism that is costing many ship owners dearly. In spite of various mechanical difficulties encountered in using superheat on locomotives, actual experience has demonstrated its superior advantages in this field.

The superheaters require no great care and attention; the principal thing is to see that the boiler is never started up with the superheaters cut out, for in that case, there being no circulation of steam through the tubes, they will burn out. Another precaution that may be taken (though it has not been found necessary) is to crack a little steam from the steam line into the superheater by valve C, Fig. 2, as soon as fires are started in the boiler. This will ensure the superheater being full of steam during the time steam is forming in the boiler. However, as there will always be some condensation left in the superheater, this will vaporize and thus fill the superheater with steam. If the superheater is not flooded from the line, the boiler stop and stop valve A in line from boiler to superheater should be opened on lighting fires to enable vapor from steam drum to get to superheater as soon as it forms and also to avoid the possibility of an unusual pressure being developed in the superheater while the steam is being gotten up and the main boiler stop valve is closed.

In order to guard against this taking place the superheater is fitted with a special relief valve set at the same pressure as the boiler safety valves. Flooding arrangements, using water, have been proposed, but these are unnecessary.

Sometimes a small pipe connection is made from the steam drum to the superheater, to ensure a supply of steam coming to the superheater as soon as steam is formed. (British practice.)

When the superheaters are fitted, attempt should be made to keep, as much as possible, a uniform temperature of the steam. What is wanted is a uniform, good, clean fire regularly fired and attended to. The admission of cold air through holes at back of fire is bad for superheat; excessive opening of furnace doors also brings down the temperature, and carrying a heavy and irregular fire will result in great variations in the temperature in the superheater.

It is thought inadvisable to go to a higher temperature than about 520 degrees F., on account of the liability of weakening valves and fittings; hence it may be advisable to

lower the working pressure below that designed in case it is found that a higher temperature than this can be maintained in the superheater.

Especially at low cruising speeds will it be advisable to work with a lower steam pressure while maintaining the 500 degrees by means of superheat. With the lower steam pressure there is less strain on boiler and piping and less leakage, and, with the temperature maintained in the superheater, the economy is just as high, or even higher, than if the regular full boiler pressure were kept at boilers.

As superheating dries the steam there is less leakage at joints, valve stems, piston rods, etc.; this also adds to the gain in economy. On the other hand, as the temperature of the steam is higher, care must be exercised that the piston-rod packing rings are of good material and that they are not fitted so tight that they may tend to bind and cut when heated up. The higher temperature introduces certain changes in condition which must be allowed for. Superheaters require little or no attention. Occasional examination and inspection of the interior of the tubes should be made.

The experience with superheaters on B. & W. boilers in the Navy in connection with the latest designs of reciprocating engines has been entirely satisfactory. The superheaters have given no trouble and have required practically no repairs. No more difficulty as to internal lubrication has been experienced with superheat than there has been without it.

### VARIOUS MATTERS CONCERNING SUPERHEATED STEAM.

A very noticeable effect of superheated steam is that it prevents initial condensation in the cylinders of the engines. Initial condensation is blamed for a large share of the cylinder losses, and it may be seen that in preventing this, superheat is accomplishing a great deal toward securing economy. Also with superheat, condensation in piping is largely prevented; this will operate to produce less leakage at joints, stuffing boxes, etc., and secure a longer life to the packing material in the joints.

On the other hand, since the temperature is considerably higher, the radiation losses will be increased and the desirability of having more efficient lagging is seen.

Steam in the superheated condition also occupies more volume than in saturated condition, and this should be allowed for in the size of piping, valve passages, etc. The increase in volume is about 12 per cent. for 100° superheat at 250 pounds.

Superheat is especially applicable to plants using reciprocating engines or impluse turbines for the initial stages.

Better conditions for the installation of superheaters can usually be secured with water-tube boilers than with Scotch boilers. In cases where uptake temperatures on Scotch boilers are relatively high, a superheater fitted in the uptake or smoke box will materially improve the economy, especially since the heat absorbed will otherwise be waste heat.

## GENERAL FEATURES OF SUPERHEATERS.

The superheater consists of a series or nests of tubes placed in the boiler and arranged so that the steam after leaving the steam drum must pass through the tubes of the superheater. The outside of the superheater tubes is subjected to the gases of combustion, and the heat from the gases heats the steam beyond the point of saturation and causes its temperature to rise above that of saturated steam. The superheater, as can be seen from the illustration, is not placed where it will come in contact with the hot gases as they leave the furnace, but is placed some distance further on, where it will come in contact with gases that have had their temperature lowered to some extent. If the gases leaving the furnace were to strike the superheating surface there would be a grave danger of burning out the superheater. The superheater should be placed in a position where the temperature of the gases is not greater than 1,500 degrees F. In the B. & W. boiler the superheater is placed above the first pass. In Scotch boilers it is usually placed in the end of the tubes or in the uptake.

*Heating Surface.*—The heating surface of the superheater in a marine boiler is from 8 to 20 per cent. of the total heating surface. Fifteen per cent. will furnish about 100 degrees superheat, with a boiler pressure of 250 pounds.

BABCOCK & WILCOX SUPERHEATER.

The superheater consists of a double row of U tubes, expanded into two cross boxes, placed at the back of the boiler, immediately above the back headers, as shown in Fig. 1. The steam from the steam drum is led to the lower

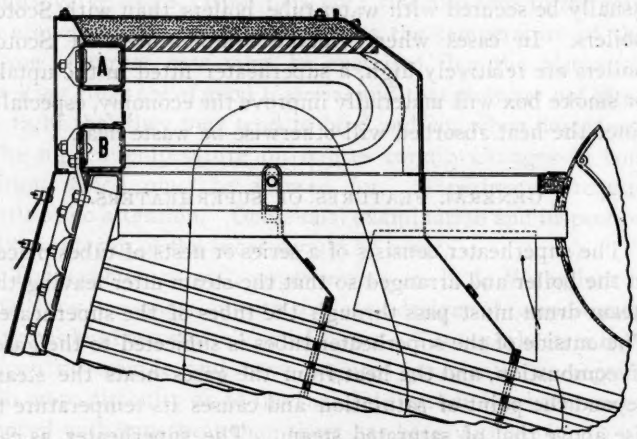



FIG. 1.—B. & W. SUPERHEATER.

cross box and the upper is connected to the steam line. The steam is thus caused to pass through the U tubes from the lower to the cross box. The superheating tubes are so placed and the baffling so arranged that the gases of combustion reach the superheater after they have made the first pass through the boiler tubes proper. While passing through the superheater the gases turn and go down to the second pass of the boiler.

The temperature of the gases at this point is from 1,100 to 1,500 degrees F. This temperature permits of securing a

fair degree of superheat without danger of burning tubes or requiring an excessive amount of superheating surface.

The U tubes are expanded into the cross boxes and hand-hole plates are fitted for each group of four tubes.

The general arrangement of cut-out valves with a B. & W. boiler is shown in Fig. 2. O is the opening to steam drum.

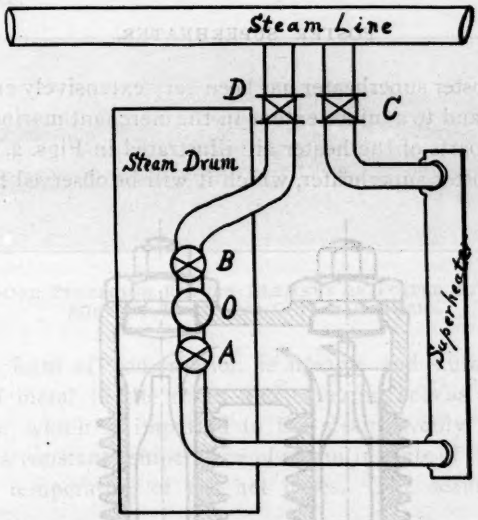



FIG. 2.—STEAM-PIPE CONNECTIONS OF A SUPERHEATER BOILER, U. S. NAVY.

Sometimes this is closed by one valve. A is a valve admitting the steam to the superheater. C is a cut-out valve in line from superheater to steam line. B is a valve admitting steam from boiler to steam line direct. D is the cut-out valve on the direct connection. Ordinarily the valves A and C are open and B and D are closed. When it is desired to reduce the superheat the valves B and D can be opened slightly, thus allowing some of the steam to by-pass the superheater. In this way the superheat can be regulated and very materially reduced without any danger of burning out the superheating surface. The double connection to the

steam line is really not necessary, since years of service have demonstrated that the superheater can be kept constantly in use. There is much less likelihood of the superheater tubes leaking than the boiler tubes proper. It would therefore be quite safe to have but one connection from the boiler through the superheater to the steam line.

## FOSTER SUPERHEATER.

The Foster superheater has been very extensively employed on shore and to a minor extent in the merchant marine. The different parts of the heater are illustrated in Figs. 2, 3 and 4.

The Foster superheater, which it will be observed from the

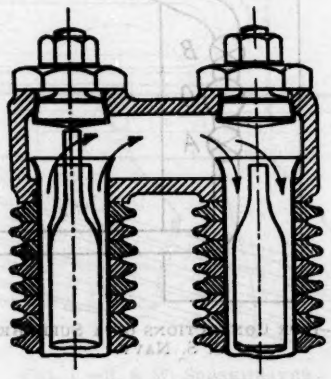


FIG. 3.—CROSS SECTION USED SHOWING METHOD OF ATTACHING RETURN HEADERS TO ELEMENTS OF FOSTER SUPERHEATERS.

illustrations, consists of a series of straight elements or tubes which are generally placed parallel to each other. The elements are joined at one end to manifolds or connecting headers, and at the other end to return headers, for which return bends are often substituted. The elements consist of straight seamless-drawn steel tubes, on the outside of which are fitted a series of cast-iron annular gills or flanges, placed close to each other and fitted to the tube so as to be practically integral with it, at the same time exposing an external surface

of cast iron, which metal is best adapted to resist the action of the heated gases. The rings or annular gills are carefully bored to gage and shrunk on the tubes. Once being in position the rings and tubes act practically as a unit. As the coefficient of expansion of steel is a trifle greater than that of cast iron, the rings grip the tubes even tighter when in service.

FIG. 4.—ONE PROTECTED RETURN ELEMENT OF FOSTER SUPERHEATER, SHOWING INLET AND OUTLET HEADERS.

This form of construction is flexible and durable. The mass of metal in the tubes and covering acts as a reservoir for heat, which is imparted to the steam evenly, tending to secure a constant temperature of steam in spite of fluctuations in the temperature of the hot gases. The seamless-drawn

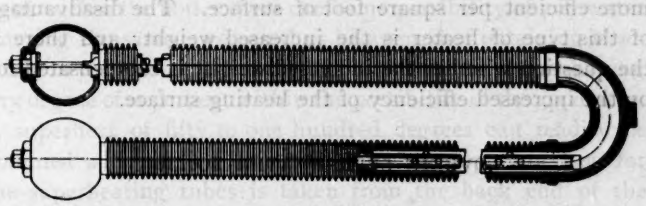


FIG. 5.—CROSS-SECTIONAL VIEW OF RETURN-BEND ELEMENT AND CONNECTING HEADERS USED IN THE CONSTRUCTION OF FOSTER SUPERHEATERS.

tube secures great initial strength which is reinforced by the rings shrunk on the outside.

*Inner Tubes.*—Inside of the elements there are placed other tubes of wrought iron, which are centrally supported by means

of knobs or buttons regularly spaced throughout their length. These inner tubes are closed at the ends. A thin annular passage for the steam is thus formed between the inner and outer tubes. The steam clinging closely to the heating surface is quickly heated in the most efficient manner.

*Joints.*—The joints at the end of the elements are made by expanding the steel tube into headers which are of wrought steel, except in special cases were cast steel is preferable.

The advantage of the lined heater is that the metal of the tubes is thoroughly guarded from burning out, and hence can be placed with safety in locations where the temperature is higher than would be safe for a plain-tube heater. The cast-iron liners perform another important office in serving as a reservoir of heat, thus securing a more uniform temperature, which is very desirable, since if the temperature does vary suddenly to any great extent there is danger of sudden expansion and contraction in the piping, cylinders, etc. This is especially important where turbines are used.

The construction of the Foster superheater is such as to make it a film heater, that is, the steam in the heater occupies the annular spaces only a fraction of an inch in thickness, while in the ordinary plain-tube superheater the steam occupies the whole area of the tube. This makes the film heater more efficient per square foot of surface. The disadvantage of this type of heater is the increased weight; and there is the question whether the increased weight is compensated for by the increased efficiency of the heating surface.

### LOVEKIN SUPERHEATER BOILER.

In the Lovekin boiler both drying and superheating tubes are located within a separate compartment formed within the boiler and extend the entire length of the boiler between the front and back heads, so that in addition to serving as drying and superheating tubes they serve as stays for the front and back heads.

They are arranged in two or more rows, one above the

other, and are staggered so as to minimize the possibility of any water passing between them. In order to insure absolutely dry steam or superheated steam being delivered from the boiler, a separate compartment is formed within the boiler. This compartment is so designed as to separate the drying tubes from the superheating tubes, so that any moisture in the steam will be removed before the steam enters the final superheating chamber. The steam enters the superheating chamber over the vertical baffle plate, passes downward over the superheating tubes and under the adjacent baffle, from where it again makes several passes over the superheating tubes before it finally leaves the boiler. It is arranged that the highest temperature of the steam meets the highest temperature of the gases within the superheating tubes, which is correct in principle. By reason of the counter-current flow of gases and steam we obtain the most efficient means of heat transference. The space for the steam passing between these superheating tubes is such that no loss in boiler pressure exists in any part of the boiler, and therefore the safety valve and the main stop valve can be fitted to a common outlet from the boiler, whereas when a dry pipe is used the safety valve requires a separate connection on the boiler. Inasmuch as the final superheating space is confined to a separate compartment within the boiler there can be no possibility of any water entering the same; we can thus be assured of obtaining absolutely dry steam or we can obtain superheated steam having any degree of superheat within the possibility of the apparatus. A superheat of fifty to one hundred degrees can readily be obtained with the boiler as shown. The supply of heat for the superheating tubes is taken from the back end of the combustion chamber through a number of tubes especially provided for this purpose, and in order to insure a higher temperature at the outlet from these tubes than that of the gases flowing through the regular tubes, the supply tubes for the superheaters are made larger in diameter. The control of the gases passing to the superheating tubes can be readily adjusted by means of a damper in the gas passage between the

supply tubes and the superheating tubes. This enables either dry steam or superheated steam to be obtained at the will of the operator.

In the single-ended Scotch boiler the back of the combustion chamber is always stayed to the back of the boiler by means of screw stays. In order to adapt this type of superheating apparatus to this type of boiler there are provided a number of small tubes at the upper part of the combustion chamber which serve as stays and at the same time permit a sufficient amount of heat to pass through them for supplying the superheating tubes within the steam space, the arrangement of the superheating tubes and the baffles within the boiler being similar to that described for the Lovekin boiler. The control of the gases passing to the superheating tubes can be readily adjusted by means of a damper in the gas passage between the supply tubes and the superheating tubes. This enables either dry steam or superheated steam to be obtained at the will of the operator.

Insulation of gas passages, also cleaning doors, are provided as shown on the plans.

### ADVANTAGES OF THE LOVEKIN SUPERHEATER FOR SCOTCH BOILERS.

1. In the Lovekin superheater the entire heating apparatus forms part of the boiler construction.

2. The steam is superheated within the boiler itself; no extra piping required.

3. The tubes can be readily and easily cleaned by steam tube cleaners.

4. The temperature of the flue gases is brought down to the least practicable limit by the use of spiral steel retarders, which are easily removed when so desired.

5. The heat for superheating is obtained direct from the combustion chamber and is controlled "as to amount" by a damper.

6. Small diameter of boiler tubes can be used to advantage so as to obtain more heating surface within a given space.

7. Simplicity and cheapness of construction is apparent from the design.

8. Smaller steam volume is permissible without any tendency to priming.

9. Choking of tubes by soot reduced to a minimum.

10. No steam joints to be broken when cleaning.

11. Efficiency must be equal to the best obtainable, as the waste gases can leave the boiler at the lowest practicable temperature consistent with that of the temperature of the steam produced.

12. Enormous heating surface can be provided within a given space on account of less steam space being required to insure dry steam.

13. Boiler can be forced to the highest limits without priming.

14. No special fittings required.

15. Every advantage in plugging, expanding or sweeping tubes, also in making internal examination of tubes.

16. Less weight of boiler and superheater combined.

In this arrangement practically no additional parts are required, and the cost of installation on the basis of power developed should be no greater than if boilers supplying saturated steam were installed. There appears to be no reason why the parts comprising the superheater should not be as durable as any other portions of the boiler.

A superheater boiler of this type is being installed on one of the tugs being built for the Navy at Seattle.

### SUPERHEATERS IN EXPRESS WATER-TUBE BOILERS.

The installation of superheaters on these boilers has been very backward, owing largely to two main causes: (1) The wrong idea that economy is of relatively less importance on vessels of the destroyer class; (2) The use of reaction turbines in connection with which superheat is not considered feasible. Since economy, in nearly all cases, carries with it a reduction in total weight of machinery and fuel, its consideration is

very important in destroyers and other fast vessels. Impulse turbines are now being applied more largely to vessels of this class, so it may be expected that superheaters will be more largely used with Express boilers in the near future. Superheaters are being used to a considerable extent with Yarrow boilers, and there appears to be no good reason why they should not be fitted on other Express boilers, such as the Normand, Thornycroft and White-Foster boilers.

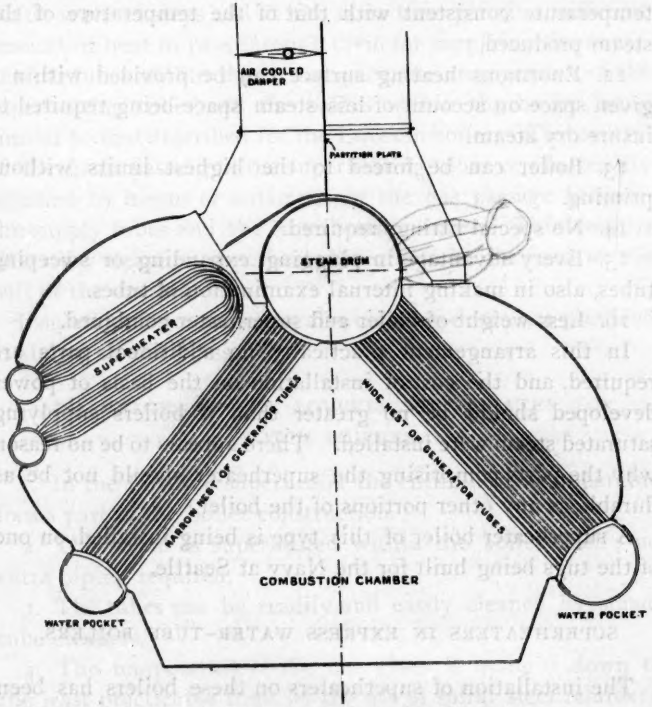



FIG. 6.—YARROW BOILER WITH SUPERHEATER.

*Superheating Exhaust or Receiver Steam.*—This can be done with a very considerable gain in economy and by utilizing what would otherwise be waste heat. If the steam from the 2d receiver of an engine or turbine be led to a

superheater in the uptake of a boiler it can absorb a large part of the waste heat, since the temperature of the steam is considerably below that of the uptake gases; furthermore 200 degrees F. of superheat might be added without the temperature of the steam becoming too high to affect lubrication or the strength of bronze or cast iron. Such a system would entail the complication of having the receiver pipes lead to the boiler and back again, but with oil fuel an alternate arrangement whereby the superheater or reheater could be placed adjacent to the engine could be adopted. For small installations the leading of the receiver pipes to and from the boiler is entirely feasible, and such an addition could readily be made to an existing installation.

This idea is somewhat exemplified by the following patent recently secured in England.

*Improvements in steam turbine installations*, Alfred Fernandez Yarrow, of Campsie Dene, Blanefield, Stirling, Scotland.

This invention relates to installations of steam-driven machinery which include as elements steam turbines and auxiliary steam engines, wherein exhaust steam from the auxiliary machinery is utilized for doing work before entering the condenser. A represents one of the boilers which provide steam for the main propelling turbines B, B, and for the auxiliary machinery of a warship or other vessel. E represents the exhaust-pipe connections from the auxiliary machinery to the steam turbines B and condensers D, $D^1$, being automatic or shut-off valves controlling admission of exhaust steam to the condenser. A heater C is provided—as shown in the drawing within the furnace casing of one of the boilers A—and

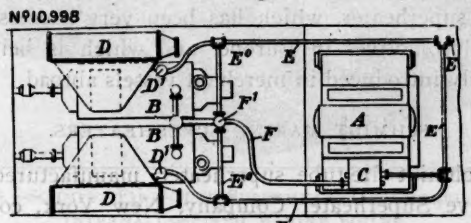



FIG. 7.

this heater is connected, on the one hand, with exhaust steam pipe E, and, on the other hand, with a pipe F leading to a valve $F^1$, and thence to the low-pressure side of the propelling turbines B, B. The valve $F^1$ controls the supply of the exhaust steam, whether heated or not, from the auxiliary machinery to the main turbines, the valve being of such a character that either heated exhaust steam only from pipe F can be delivered to the main turbines from the auxiliary machinery, or the exhaust steam from the auxiliary machinery can be delivered through pipes E, $E^0$ to the main turbines without passing through the heater, or a mixture of heated and non-heated exhaust steam from pipes F, E, $E^0$ can be delivered to turbines.—February 25th, 1914.

The above arrangement could readily be applied and it would secure additional economy, whether auxiliary exhaust were used for feed heating or not. On oil-burning vessels blowers are required in any case to secure the draft; hence the temperature of the smoke pipe might be reduced to any extent without affecting the draft. The possibility of securing as much as 10 or 12 per cent. reduction in steam consumption is here presented.

### OTHER TYPES.

In Scotch boilers superheaters are sometimes fitted by putting a U tube into the boiler tubes at the smoke-box end. The U tubes are connected to steam headers which act as steam receivers. The presence of these U tubes in the boiler tube no doubt causes soot to collect more readily and makes it somewhat more difficult to clean these tubes. A type used to a considerable extent with Scotch boilers abroad is the Schmidt superheater, which has been very successfully applied to locomotives in Europe, and which is being quite extensively introduced in merchant vessels abroad.

### SCHMIDT MARINE SUPERHEATERS.

The Schmidt fire-tube superheater, manufactured by the Locomotive Superheater Company, New York, consists of collector castings and a system of units or elements made up

of U-bent tubes, the material of which is cold-drawn seamless steel. The collector castings are located in either a vertical or horizontal position in the uptake end of the boiler. The units, which are arranged in groups leading in and out of the uptake end of the flues, are expanded in the flanges or collars, which in turn are fastened to the collector castings. In joining the ends of the unit pipes to the collector castings one end of the pipe is in communication with the header from the boiler and the other with the steam pipe leading to the engines. Thus the steam in passing from the boiler to the engines must pass through the units or elements in the tubes, where the superheating takes place. The connection between the units or elements and the collector castings is made by means of a single clamp using but one bolt or stud, thus facilitating the removal of the units should occasion demand their removal.

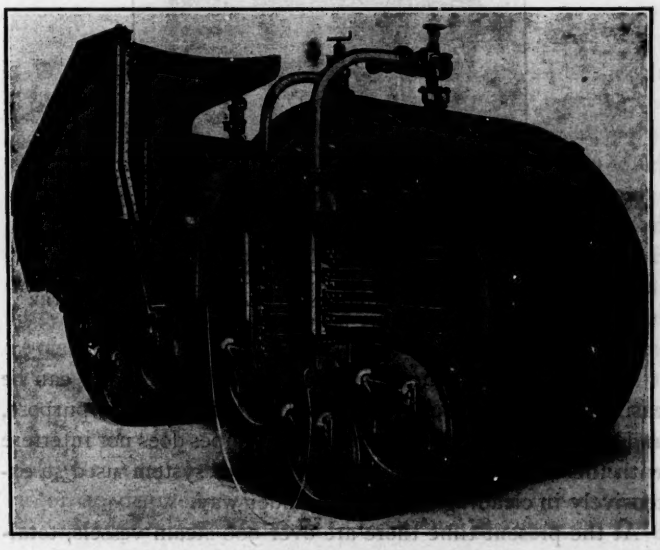


FIG. 8.—SCHMIDT FIRE-TUBE SUPERHEATER INSTALLED IN AN INTERNALLY-FIRED MARINE BOILER.

The units are formed by welding the straight sections of the pipe to a forged return bend, whose thickness is somewhat greater than the thickness of tube. By this form of construction each unit consists of a single continuous pipe and does away with screwed joints or connections which might be contributory to leaks. The construction of the Schmidt superheater is evident from Figs. 1 and 2, which show its application to an internally-fired Scotch marine boiler.

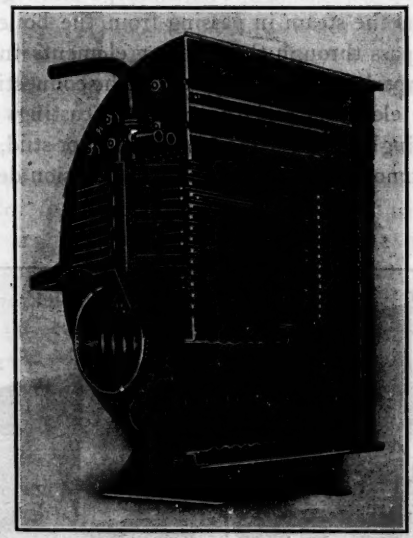


FIG. 9.—SECTIONAL VIEW, SHOWING LOCATION OF SUPERHEATER ELEMENTS IN FIRE TUBES.

The cleaning of the superheater and boiler tubes can be easily effected by steam jets specially provided for the purpose, and the introduction of the superheater pipes does not interfere with the well-known "Diamond" blower system used so extensively in cleaning flues while under way.

At the present time there are over 900 steam vessels, totaling over 1,000,000 horsepower, equipped with Schmidt marine superheaters.

The most economical results are obtained when a steam temperature of 600 degrees F. is employed. It is adaptable to either new or existing boilers of the fire-tube type and can be applied with no change in design or construction. Its construction provides easy access to all screwed joints for ready removal of same.

### OIL FUEL AND SUPERHEAT.

The advent of oil fuel should be favorable to the introduction of superheaters on marine boilers. With oil fuel an even and more regulated temperature of furnace can be maintained, and this operates to secure a better average degree of superheat. With oil fuel there is less manual labor required and the boiler attendants are likely to have more mechanical and engineering knowledge, and will, for this reason, be likely to take more care in securing better operating conditions for the superheater.

### SUPERHEATERS IN UPTAKES.

With Scotch boilers superheaters are sometimes installed in the uptakes, where the heating surface is in contact with the gases after leaving the boiler. If the boiler is efficiently proportioned the temperature of these gases should not be much over 500 degrees F. This temperature does not permit of giving any considerable amount of superheat. If the steam pressure is 200 pounds its temperature will be 388 degrees F. So that with the gases at 500 degrees probably not more than forty degrees of superheat will be obtained. If the superheater is placed in the path of the gases somewhere where they can give up their heat to the superheating surfaces before they are clear of the boiler, a much greater degree of superheat may be secured. The boiler-heating surface may also be so proportioned that the gases, on finally leaving the boiler, are but 30 or 40 degrees above the temperature of the steam in the boiler proper, and considerably below the temperature of the steam as it leaves the superheater.

If the best economy is desired the uptake cannot be con-

sidered the ideal location for the superheater. But if the boiler as arranged does not produce a low uptake temperature, a superheater installed in the uptake may secure a very considerable gain in economy, since it will operate on waste heat.

Air heaters and flue-gas feed heaters have a logical place in the uptakes, and because they operate on lower temperatures they are able to abstract more of the heat than it is possible for a superheater to do.

The types of superheaters installed in uptakes of marine boilers are, "Hides," in England, and the "Foster" superheater, in this country. The Foster superheater was so installed on the steamship *Brazos*, operated very satisfactorily, and assisted in giving this vessel a record for fuel economy.

*Flue-gas feed heaters* are again being proposed abroad, especially in connection with Express boilers. A boiler built for geatest economy would have a superheater, a flue-gas feed heater and also an air-heating apparatus similar to Howden's system. This may appear complicated, but if carefully designed would not be cumbersome. With these installations on an Express boiler the temperature at base of stack could be kept down to 400 degrees F., the air entering the furnace delivered at a temperature of 180 degrees and the feed raised 40 degrees above that delivered by the exhaust-feed heater.

By means of the flue-gas heater, or economizer and the air-heating apparatus, the ten to fifteen per cent. of heat lost in the chimney gases can be reduced to six or seven per cent.

### SEPARATELY-FIRED SUPERHEATERS.

Separately-fired superheaters have been used to some extent abroad, especially in Germany. In this case the superheater is built very much like an Express boiler, but saturated steam instead of water is delivered to it. It would appear that there would be considerable danger of burning out the tubes of such a superheater, since here the furnace gases will act directly on the steam-contained tubes without being previously cooled by water-carrying tubes. Also since the temperature

of the steam is higher the temperature of the gases leaving the superheater will be considerably higher than would be necessary in a boiler. Therefore, if no economizer is fitted, there will be more waste in the uptake gases than if each boiler contained its own superheater, placed intermediately in the path of the gases. It would appear that a separately-fired superheater would add greatly to the complication, and its advantages over the attached superheater are not apparent.

## GENERAL REMARKS.

Superheating has not made the progress in the marine service that its undoubted advantages should cause it to do. This has been due to conservatism and an apparent fear of trying something new. In shore practice, especially in large electric-power plants, superheating is the rule. In marine work, where economy of fuel is even more important, it has been largely neglected.

Superheating up to a temperature of 500 degrees F. certainly creates no additional difficulty. For higher temperatures special provisions will have to be made, and it is believed that provision for using higher temperatures will be justified.

With a few exceptions, superheating has not been used with Parsons marine turbines, owing to the fear that the larger variations in temperature may cause some unusual expansion and possibly produce blading troubles. It cannot be said that this fear is entirely justified, since superheat has been used with Parsons marine turbines without causing any trouble. Superheat has been used with the Curtis marine turbine and with both the Curtis and the Westinghouse turbines on shore. With impulse blading at the high-pressure end there should be no cause for blading troubles due to superheat, and it would appear that one of the chief advantages of impulse blading is that it permits the use of higher temperatures and allows of superheat being used under more favorable conditions.

## EFFECT OF HIGH VACUUM.

The securing of a high vacuum is a most feasible method of improving economy of steam consumption. By the utilization of a high vacuum more heat units can be taken away from the condenser losses and made to do useful work. The amount of gain by having a better vacuum, and the relatively slight cost and trouble to secure it, are not generally appreciated in marine work. Operating engineers are altogether too easily satisfied on this point. Usually a vacuum of 25 inches of mercury, or within 5 inches of the barometer, is considered good enough, when by a little care and attention perhaps 2 inches more could be secured.

We are now working with the opposite end of the problem from that considered in connection with superheating. In superheating it was a proposition of increasing the initial temperature; now it is a problem of reducing the final temperature. The formula $\frac{T_1 - T_2}{T_0}$ still comes into play. If a vacuum of 27 inches is carried, $T_2$ will be 112. If a vacuum of 29 inches is carried $T_2$ will be 76. This difference will increase the theoretical economy over 12 per cent. where the initial pressure is 250 pounds, or would have the same effect as an increase of pressure from 250 pounds to 370 pounds.

The practical gain is not as great as this, due chiefly to defective arrangements in the engines' increasing carry-over losses which do not allow the high vacuum to be properly used. It is more or less generally said that the reciprocating engine cannot use a high vacuum to advantage. This statement is not really borne out in theory or in practice. A better vacuum is always some gain, no matter what kind of an engine is used. It is only when means for securing the higher vacuum call for a great expenditure of weight or power that the limiting vacuum that is desirable to carry is reached. The greater the temperature range, other things being equal, the greater the economy, and in any existing installation the highest possible vacuum practicable should be carried.

It has been claimed that marine reciprocating engines secure little or no economy by carrying a vacuum higher than 24 inches in the L.P. cylinder. There is not known of any conclusive data to substantiate such a claim and, though definite experiments have not been made, the observation on many engines has failed to show any case where a better vacuum did not improve the economy. Theoretically there is an enormous gain, and unless extraordinary conditions prevail a good portion of this gain can be realized in practice. With turbines the gain due to the higher vacuum is much larger than with reciprocating engines, and every effort to secure a vacuum of at least 28 inches in the condenser should be made.

In any case the engineer operating a plant will invariably find it to his advantage to obtain the best vacuum that he can get, and the old practice, sometimes indulged in, of deliberately lowering the vacuum to increase the temperature of air-pump discharge, is certainly working in the wrong direction.

The theoretical gain is clearly shown by the steam-expansion curve, and any increase in vacuum increases the possible work that can be done.

The vacuum that it is possible to carry is limited by the temperature of the circulating water. Sea water is hardly ever above 85 degrees F. The vapor pressure corresponding to this temperature is about 28.7 inches of mercury. With a temperature of 60 degrees F. it is 29.40 inches, so that, leaving a half an inch leeway, a vacuum of 28 inches can be secured in the tropics and 29 inches where the sea water is below 60 degrees F. Most vessels carry fully an inch less than this, which means that they are needlessly wasting from 1 to 5 per cent. of their fuel. There is no good practical excuse for a vacuum of less than 27–28 inches in marine work.

Of course the air pump and condensing apparatus fitted may not be able to deliver this vacuum and the design of engine and exhaust pipe may be such that the high vacuum cannot be fully utilized. But condensing apparatus that will

give the vacuum and engines that will fully utilize it can be and are being built, and they will cost but little more than the old apparatus which, for the ordinary vessel, allows 5 per cent. of available heat of the steam to be wasted. The necessary increase in weight is almost *nil*.

### AIR AS AFFECTING VACUUM.

The maintenance of a high vacuum is mainly a question of excluding air from the condenser. Even with bad condenser design the steam will be condensed if enough circulating water and enough surface is supplied. The first thing, then, and by far the most essential in attacking this problem, is to *exclude the air*.

Some air is contained in the steam as it leaves the boiler, but by far the greater share that finds its way into the condenser is leakage into the engine and exhaust system.

The air in the steam itself comes in largely with the feed water; hence to reduce this air the following can be done:

(1) Provide feed tanks of large liberating area where the air may, as far as possible, be liberated. Avoid churning and agitating the water in the feed tank as much as possible.

(2) Liberate air from feed on delivery from feed heater. As the air is heated in the feed heater it will have a tendency to be driven off, and if an air-collecting chamber and blow-off cock are provided on the discharge side of feed heater a considerable amount of this air may be got rid of.

(3) In the steam space of the boiler the air will collect immediately above the water line, so if the surface blow is used occasionally when the water level is just below level of scum pan some of this air may be driven off.

*Air leakage into the system.*—Air, of course, can only come into the system at points where the pressure is below the atmosphere. All exhaust joints, valve stems, glands, etc., are possible sources of air leakage.

In reciprocating engines the L.P. piston and valve rods, and all connections, such as drains, relief valves, etc., are pos-

sible places where air may be taken in. Relief and drain cocks can be made tight, but the sliding joint of the piston and valve-stem stuffing boxes are difficult to make absolutely airtight. Air, however, may be excluded from these joints by making them steam packed; that is, a steam gland is formed and a little steam at slightly above atmospheric pressure is admitted from some source such as the auxiliary exhaust line. Marine steam-turbine glands and expansion joints under a vacuum are packed with steam to prevent air leakage.

Most of the exhaust joints are likely to be connected with drains. If these drains are led to the main condenser any air leakage will affect the vacuum. If these drains can be led to the feed tank or to the auxiliary condenser the air that may be drawn in will not affect the vacuum in the main condenser. *Consequently*, drains from traps, feed heaters, evaporators, cylinders of auxiliaries, etc., should be discharged to the feed tank, and not to the main condenser, as they often are with no thought given to the question of their enabling large quantities of air to be admitted to the condenser.

If the auxiliary exhaust is led to a feed heater and this heater has no efficient air-discharging device, the air will, from time to time, be discharged through the feed-heater drain. If this leads to the main condenser this air will impair the vacuum.

The auxiliary exhaust lines have numerous joints that are not always tight, so that if the auxiliary exhaust line is kept at a pressure below the atmosphere air can leak in. Therefore the auxiliary exhaust pressure should be kept above the atmosphere so that the air may be excluded and kept out of the system.

The auxiliary exhaust pipe will have numerous sources of air leakage, and if this exhaust is thrown into main condenser the air will also come in. It is better to utilize the auxiliary exhaust as far as possible in the feed heater where, if the drain leads to the feed tank, the air can not get to the main condenser and impair the vacuum.

It will thus be seen that there are numerous sources of air

leakage that can be closed off without any inconvenience. Just stopping these leaks may mean an inch difference in the vacuum on the main engine.

Now in practice the increase in vacuum that can be secured by care in closing off all avoidable air leakage is greater by far than any increase that may be secured by the utmost refinements of condenser and air-pump design. Any perceptible drop in vacuum from that corresponding to the temperature of the discharge water is, in nine cases out of ten, due to some air leak. The remedy is to *find the leak and stop it.*

*Air in the Condenser.*—Having by every means possible excluded and prevented the admission of air into the condenser, the next thing is to find means for removing that which does gain entrance.

The bad effect of air in the condenser is caused by its presence rendering the cooling surface inactive, since no steam can occupy the space taken up by the air. Its presence also causes an increase in the pressure, since the air, unlike the steam, can not be condensed.

Air will collect above the water, so that to reach the air effectively it is necessary to have an air suction separate from the suction which takes away the condensed water.

Until the advent of the turbine all marine condensing plants simply had an air-pump suction at the bottom of the condenser and the matter of air in the condenser was not materially considered. If trouble was experienced more cooling surface or a larger air pump was supplied. The turbine brought into marine practice a more thorough consideration of condensing apparatus by which improved apparatus, represented by the augmentor, dual air pumps and various dry-vacuum systems, were developed.

### TREATMENT OF AIR IN CONDENSERS.

The greater the density of the air in the condenser the easier may it be handled, but with very high vacua the density is very low and the volume occupied correspondingly increased,

hence the higher the vacuum the more difficult does the problem of handling the air become. One method of increasing its density is, however, at hand, and that is to cool the air. By cooling the air its volume is decreased and it can be more readily handled. The air can, of course, be cooled by increasing the volume of circulating water, but in doing this the condensed water also is cooled and heat units are lost.

If, however, the air and the water can be separated and the air then cooled separately, the water may be retained at the temperature corresponding to the vapor pressure, while the air is cooled as much as possible and the ability of the air pump to handle it is increased.

This matter of taking the air off separately and cooling it is the essential point in the new systems of condensing apparatus.

Different systems of doing this have been devised and put in operation on marine work. A short reference to several systems will be made. Complete descriptions of several systems may be formed in the references.

### BAFFLING.—KINETIC EFFECT.

The question of baffling a condenser is one about which many arguments have been advanced, but of late years some real basic principles on this point have been brought out and made use of practically.

In general such baffling as creates an obstruction to the free flow of steam is objectionable. If the surfaces are properly arranged and the exhaust connections properly led, so that all portions of the tube surface receive their proper share of the exhaust, baffling is not necessary. Properly-designed baffling may assist the proper flow of steam. Improperly-arranged baffling will disturb the flow of steam and create resistance.

Baffling is associated with the matter of the area of steam space at various parts of a condenser. The sole reason for baffling is to guide and assist the flow of exhaust from the

exhaust entrance to the outlet for condensed water and air. The nature of the exhaust duct will consideraby affect the baffling of a condenser. If it can be so arranged the exhaust should enter the full length of the tubes. This should be done; if it cannot be done, some arrangement of scattering plate or baffling must be provided to guide the exhaust proportionately to the different parts of the tube surface.

With turbines the exhaust duct can usually be constructed so that the entrance covers nearly the full length of the tubes.

In a reciprocating engine, especially with built-in condensers, the exhaust pipe connects at some point at the top of the condenser shell, usually at one end. This is not a good arrangement, and it would be much better to lead the exhaust so that it enters near the middle of the length. When the exhaust enters at one point baffling should be provided to cause the steam to distribute itself toward the ends of the condenser.

The object should be to have the least possible obstruction and difference of pressure from *a* to *o* in the diagram, Fig. 12a; any increase in this resistance is so much actual increase in back pressure. The presence of such baffles as are indicated by *b* is not believed to be essential, and there is some question concerning the lower baffle, though this latter is quite extensively employed, its object being to assist in the separation of the air.

*Volume of Steam Space.*—There must be sufficient volume or area of cross section of flow at all points so that no restriction is created. This is even more important with a reciprocating engine than in a turbine. In order to secure proper passage for the steam some of the tubes are left out in the upper part of the condenser and channels to facilitate the flow of steam among the tubes are thus created.

A reciprocating engine exhausts only about for one-fifth of the stroke, hence there is a pulsation—a rise and fall of pressure in the condenser. Suppose at the end of the period of closed exhaust the vapor pressure is one pound absolute and that at each exhaust opening one-half of the volume of exhaust space of steam is injected into the condenser. The pressure just

before closing exhaust would then be somewhere near 1.5 pounds absolute and the average back pressure would be approximately 1.25 pounds absolute. Now, if the exhaust were continuous, as with a turbine, the average back pressure would be 1 pound absolute. But by increasing the volume of the exhaust space to double what it was, the volume of steam

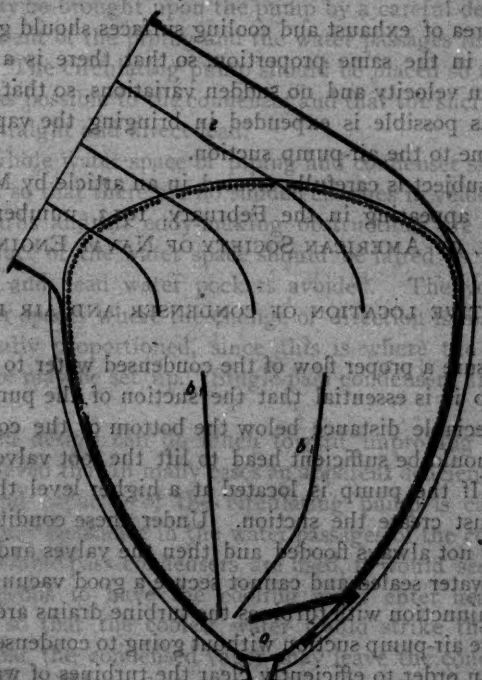


FIG. 124.

coming in at each exhaust opening would only be one-fourth instead of one-half the exhaust volume, and the back pressure near end of exhaust would be 1.25 pounds instead of 1.5 pounds absolute. The average back pressure would be 1.125 pounds absolute, instead of 1.25 pounds, and by thus increasing the exhaust volume the back pressure would be reduced a

quarter of a pound. With turbines there is a constant exhaust and the vapor path is preferably as short as possible.

The steam as it enters the condenser should come into contact with the greatest possible cooling surface. This naturally points to condensers of oval or wedge-shape or to tube spacing in cylindrical condensers that produces the same results.

The area of exhaust and cooling surfaces should gradually decrease in the same proportion, so that there is a gradual change in velocity and no sudden variations, so that as little energy as possible is expended in bringing the vapor from the engine to the air-pump suction.

This subject is carefully treated in an article by Mr. D. B. Morison appearing in the February, 1913, number of the JOURNAL OF AMERICAN SOCIETY OF NAVAL ENGINEERS.

## RELATIVE LOCATION OF CONDENSER AND AIR PUMP.

To insure a proper flow of the condensed water to the wet-air pump it is essential that the suction of the pump be at an appreciable distance below the bottom of the condenser. There should be sufficient head to lift the foot valves of the pump. If the pump is located at a higher level the pump itself must create the suction. Under these conditions the pump is not always flooded and then the valves and pistons are not water sealed and cannot secure a good vacuum.

In conjunction with turbines the turbine drains are usually led to the air-pump suction without going to condenser. This is done in order to efficiently clear the turbines of water, and when the drains are so connected the air-pump suction must be several feet below the lowest part of the turbine casing.

If a dry and a wet-air pump are supplied, the wet pump must be in a low position to secure proper flow of water to it, and though the dry pump may be somewhat higher, it is believed better results will be secured if the dry pump is at a lower level than the condenser.

### THE CIRCULATING WATER.

The main question with the circulating water is to get enough of it in the right place. It is, of course, essential to have a circulating pump that is economical and whose cost and weight is within proper limits. As the lift is low, the centrifugal pump is almost invariably used. A great deal less work may be brought upon the pump by a careful design and arrangement of the piping and the water passages in the condenser. The circulating pump should be placed so that it is as near as possible to the condenser and that the suction pipes have a straight and direct lead.

The whole water space of piping and condenser should be designed so that there are no sudden changes in velocity, that sharp turns and all eddy-making obstructions are avoided. The surface of the water space should be faired as much as possible, and dead water pockets avoided. The volume of the water spaces where the change of direction is made must be carefully proportioned, since this is where the greatest resistance may be set up. Single-pass condensers are preferable.

Care in design can do much toward improving matters. It is believed that in many cases an apparent deficiency in the power and capacity of the circulating pump is caused by unnecessary resistance in the water passages of the condenser. Where double-pass condensers are used, it would seem more advantageous to have the cooling water enter nearest the exhaust so that the coolest water would strike the hottest vapor, that the condensed vapor may leave the condenser as warm as possible, and that the steam may be condensed as near as possible to the engine. This is, however, still a debatable matter.

### CONDENSING SURFACE REQUIRED.

The amount of condensing surface required can be materially reduced by greater care in design so that all portions of the surface are fully effective. In past years it was the

practice to allow a certain amount of surface per I.H.P. In general, more surface than necessary was provided. The surface needed will depend upon the water consumption of the engine, the exhaust pressure and the temperature of the circulating water. A deficiency of surface can, however, be made up by an increase in the amount of circulating water.

With sea water at 60 degrees F. about 40 pounds per square foot of surface is about the maximum amount of exhaust that can be condensed. This, however, requires a circulating-water velocity that it may be undesirable to maintain. A further point to note is that the better the vacuum the less heat will there be for the circulating water to take away. Hence, paradoxical as it may seem, less circulating water will be required the higher the vacuum is carried. The allowance of surface and circulating water must, however, be adequate when the sea temperature is as high as 80 degrees F. Hence one square foot of surface should be allowed for 20 pounds of water condensed.

Some of the best types of marine condensing apparatus for efficiency, lightness and generally satisfactory operation are found in some of our present destroyers. These are all single-pass condensers, and the expanded tubes are employed. The condenser of the *Cassin* is illustrated in Fig. 12b; that of the *Duncan* in the article describing trials of that vessel in this number.

The following results have been obtained on some recent destroyers:

| Water condensed per hour per sq. ft. cooling surface. | Temperature of circulating water, degrees F. | Vacuum. |
|---|---|---|
| 21.2 | 55.0 | 27.8 |
| 27.08 | 56.9 | 27.87 |
| 22.2 | 60.8 | 28.47 |
| 20.6 | 59.4 | 28.24 |
| 23.78 | 64.3 | 27.5 |

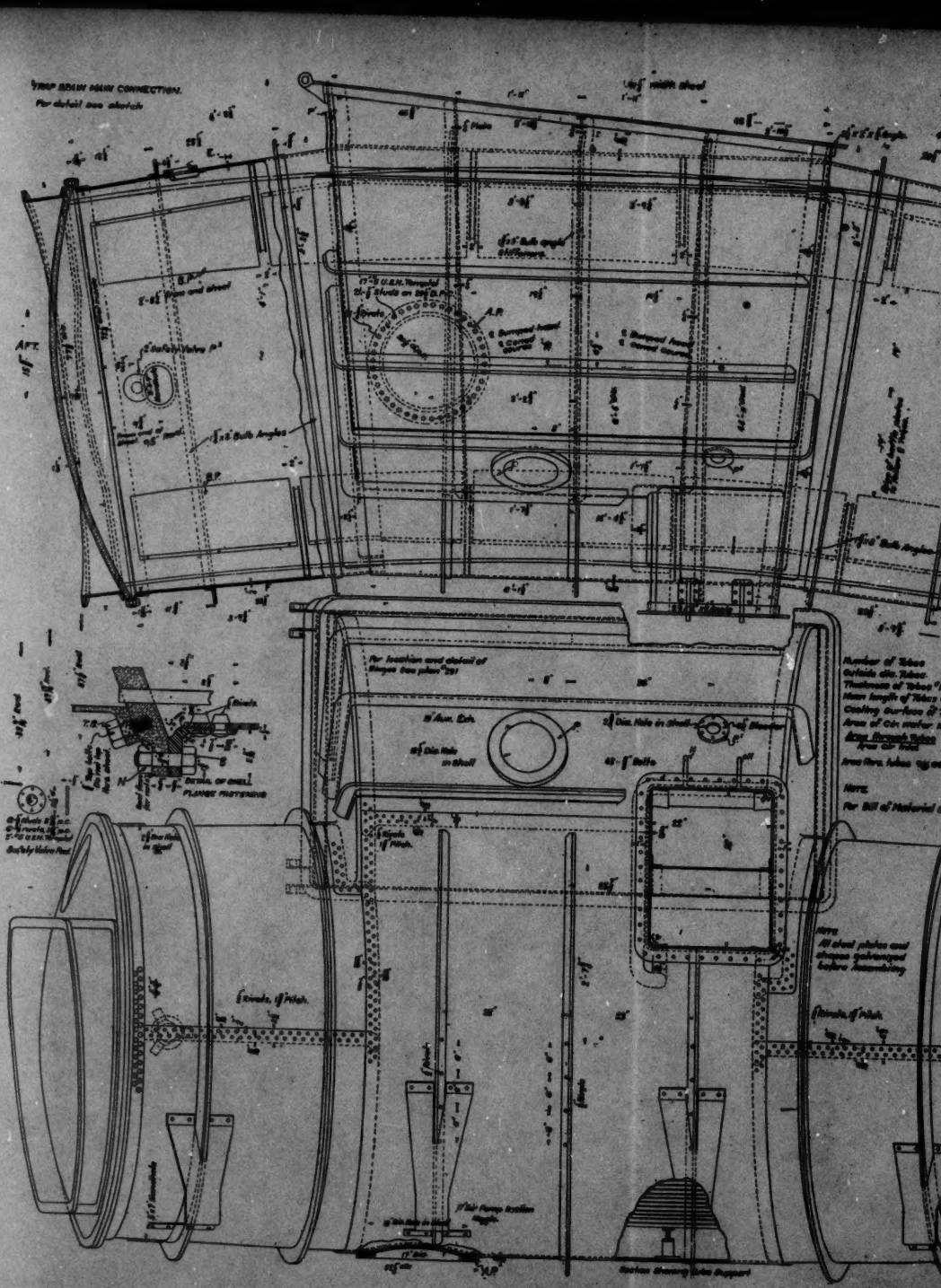

TRAP DRAIN MAIN CONNECTION.
For detail see sketch
AFT.
FWD.
Number of Tubes
Outside dia. Tubes
Thickness of Tubes
Mean length of Tubes
Cooling Surface of
Area of Air water
Area Steam Space
Area Car Tanks
NOTE
For Bill of Material
NOTE
All steel plates and
shapes galvanized
before Assembling
Section Showing Tube Support

FIG. 12B.

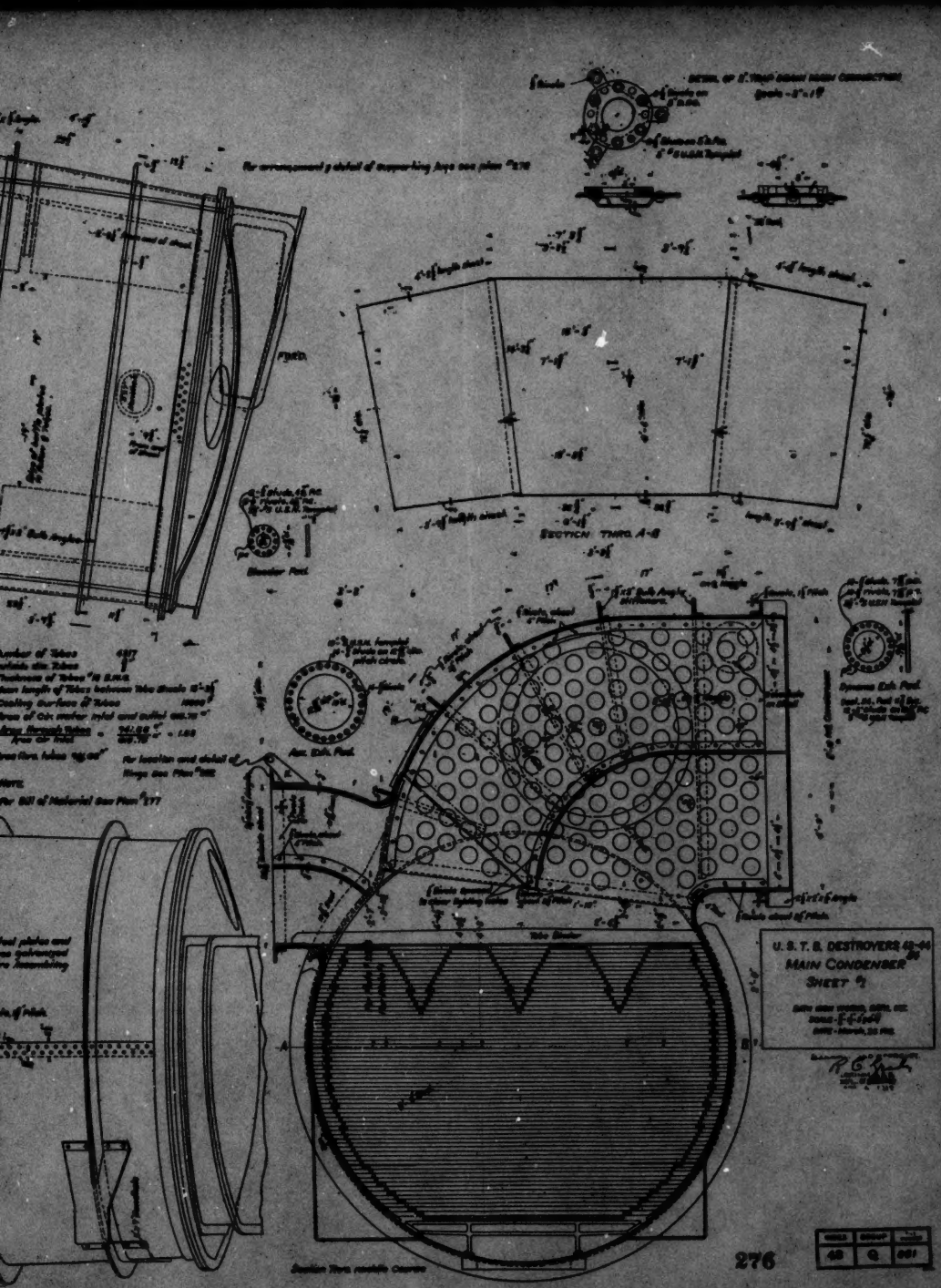

U. S. T. B. DESTROYERS 43-44
MAIN CONDENSER
SHEET 9
SECTION THRO. A-B
276

FIG. 12B.

These are actual results in which the water was actually measured. They show a condensing capacity greatly in excess of what has usually been allowed in marine practice. It is probable that a considerably greater capacity could be secured with the same condensers. The results above are those secured under full power. These results indicate that the double-pass condensers used on large vessels have considerably larger surface than should be necessary or that they are considerably less efficient than the condensers installed on the destroyers. Single-pass condensers appear to be gaining in favor, and a very large number of Weir uniflex condensers employing the single-pass have been installed in the last few years. This type of condenser is employed on the Argentine battleships built in this country.

The single-pass offers much less resistance, and hence in reality increases the capacity of the circulating pump. Since increased pump capacity can be obtained on less weight than increased cooling surface, it would appear rather more desirable to provide excess of pump capacity than excess of condensing surface.

*Type of Drive for Circulating Pump.*—As the speed of the pump has to be greatly varied to suit different conditions of speed and temperature of water, and as at times it may be desired to run at relatively low speeds, it appears that a reciprocating engine with forced lubrication would be most suitable. A *small turbine* can be used, but, owing to variations in speed, this will probably be more wasteful of steam.

*Electric motors* may also be used, but they are not as susceptible to speed variations, and electric apparatus cannot be well taken care of in places where the circulating pump is usually installed.

### WESTINGHOUSE MARINE-SERVICE CONDENSERS.

The development of the Westinghouse marine-service condenser is coincident with the development of the Westinghouse marine turbine. The particular construction embodied in this

turbine enables the condenser to be placed directly beneath the turbine. The turbine is short, the exhaust covers the whole bottom of the casing, so that essentially the exhaust opening distributes steam through practically all portions of the condensing outfit. The placing of the condenser beneath the turbine necessitates the design of a shallow condenser. This, of course, is necessary so that the centerline of the turbine shaft is not too high. However, this does not present a great deal of difficulty, since the reduction gear is generally arranged with a pinion on top, so that the center of the turbine shaft can be materially higher than the axis of the propeller shaft. In other words, the condenser design is that comprising a condenser of small vertical height as compared with conventional designs, but, due to the fact that the pinion can be placed on top of the gear, the vertical height is sufficient if the condenser is properly designed.

With the condenser of small vertical height there are some problems that come in to so proportion it that there will not be selective flow through some portions of the tubing. Where the distance from the top to the bottom of the condenser is very small there will be very little pressure drop or difference in vacuum between the top and the bottom. In other words, but a small difference in pressure is necessary to create flow from the upper regions of the condenser to the lower regions of the condenser. This means that those portions that are benefited by the velocity of the steam coming from the turbine wheel will tend to have increased flow, while those portions situated some distance from the turbine wheel will have decreased flow, due to the fact that there is not any help given to the flow by the residual velocity of the steam. Special baffling, determined by experiment and calculation, has very successfully worked out this part of the problem. The shallow condensers as made have a fairly uniform flow, thus maintaining the maximum condenser efficiency. It is of interest to note that one portion of this design lends itself to great efficiency, and that is the small loss in vacuum due to friction in traversing the tubes. The distance is so short that this loss

is negligible. The vacuum in the turbine exhaust is generally less than .1 inch lower than that existing at the air pump. This, of course, is much better than that usually gotten with the conventional type of condenser where the length of flow from the upper portion of the condenser to the lower portion is long. Fig. 13 shows a cross section of the condenser. The small vertical depth is at once apparent. Fig. 14 shows the

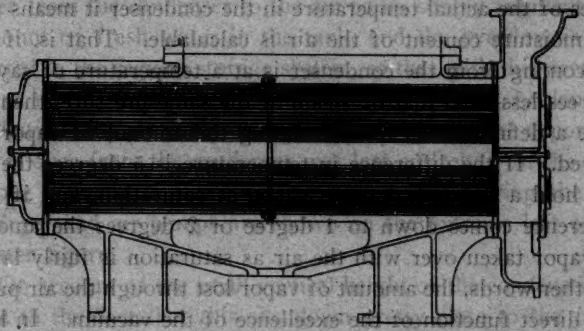


FIG. 13.—WESTINGHOUSE CONDENSER.

tube plate, giving an idea of the tube distribution in a section cut at right angles to the tubes. Baffling, in order to create uniform flow through the various zones of the condenser, is provided.

The most striking portion of the Westinghouse condenser

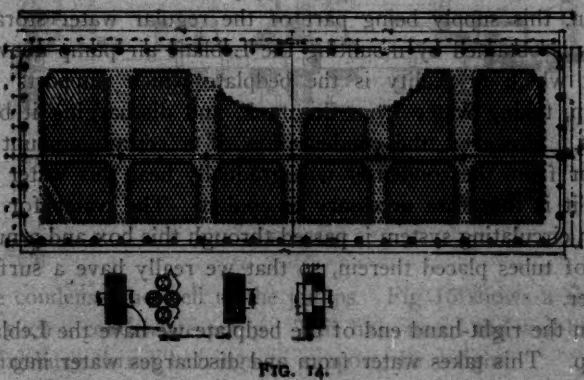


FIG. 14.

system is the Leblanc air pump (fully described in the November, 1912, JOURNAL AMERICAN SOCIETY OF NAVAL ENGINEERS). This pump can maintain a higher vacuum than most any other type of air pump commonly in use. The very fact, however, of its high vacuum-producing ability brings other problems to the front to be dealt with. For example: if the air pump can maintain a vacuum whose temperature is within 10 degrees of the actual temperature in the condenser it means that the moisture content of the air is calculable. That is, if the air coming from the condenser is at a temperature of say 10 degrees less than that of steam at the same pressure, then we have a definite basis for calculating the amount of vapor removed. If the difference in temperature is 5 degrees the air can hold a great deal more vapor at saturation, and if this difference comes down to 1 degree or 2 degrees the amount of vapor taken over with the air as saturation is fairly large. In other words, the amount of vapor lost through the air pump is a direct function of the excellence of the vacuum. In land work this is not of extreme importance. That is, if 100,000 pounds of steam per hour are being condensed, the loss of 500 pounds per hour through the air pump does not mean very much, but on a marine installation the continued loss of such an amount of vapor would be serious. It is therefore necessary and expedient to arrange the Leblanc air pump for marine work so that it uses the same supply of water over and over again, this supply being part of the regular water storage. This is obtained by mounting the Leblanc air pump above a tank which in reality is the bedplate which supports the pump, taking water from this tank, and discharging it back to it. However, the condensation of a certain amount of vapor from the condenser will gradually heat this water up, and it is therefore necessary to cool it. The water for the main circulating system is passed through this box and a number of tubes placed therein, so that we really have a surface cooler.

On the right-hand end of the bedplate we have the Leblanc pump. This takes water from and discharges water into the

bedplate tank. It circulates from one end of this tank to the other, and in doing so passes through the tubes shown about the center of same. The main circulating pump takes its suction through the bottom of the bedplate, and water passes around the outside of the cooling tubes, so that the temperature of the fresh water in the tank is kept substantially at that of the circulating water. In this way the vapor that comes over from the condenser to the air pump simply augments the amount of fresh water in the tank, and the overflow is connected to the regular storage system, so that with this arrangement no vapor is lost. On some of the earlier Leblanc installations the air pump used salt water, so that the vapor that came over from the condenser was lost entirely. In the print above referred to the gear and turbine for driving the centrifugal and air pumps are shown on the extreme left of the center figure.

On account of the small vertical height of the condenser and due to the fact that this sits down pretty close to the frame of the ship, it is necessary to use vertical condensate pumps. This is to obtain the location of the condensate pump at the lowest possible point, so that there will be sufficient fall from

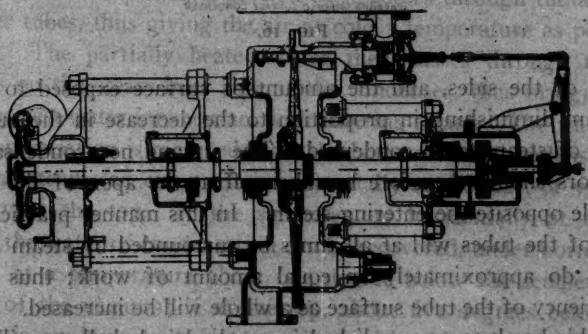


FIG. 15.

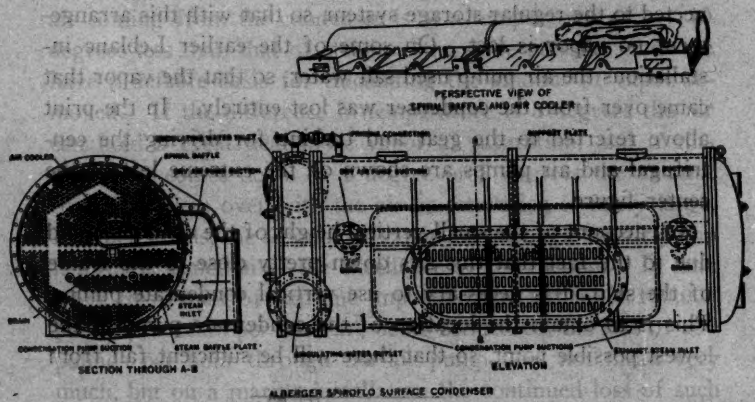
the condenser hotwell to the pumps. Fig 15 shows a section of such a condensate pump. This has one of the well-known Kingsbury thrust bearings to support the weight.

## SPIROFLO SURFACE CONDENSER.

The latest *Alberger system* employs a spiroflo surface condenser, a rotating dry-air pump, or a hydroflo centrifugal dry-air pump and a centrifugal condensation or hotwell pump.

These parts are described as follows:

This condenser is shown in illustrations, Fig. 16. The theoretically correct form for a surface condenser is one having a triangular cross section, the steam being admitted over




ALBERGER SPIROFLO SURFACE CONDENSER

FIG. 16.

one of the sides, and the amount of surface exposed to the steam diminishing in proportion to the decrease in the quantity of steam to be condensed. The air and non-condensable vapors should therefore be taken off at the apex of the triangle opposite the entering steam. In this manner practically all of the tubes will at all times be surrounded by steam and will do approximately an equal amount of work; thus the efficiency of the tube surface as a whole will be increased. All of this has been accomplished in a cylindrical shell, as will be seen by referring to the sectional cut, showing the *spiroflo* surface condenser.

The steam enters tangentially and is distributed over the full length of the tubes by a rectangular steam dome, a perforated

baffle plate being provided opposite the exhaust inlet. Large steam bleeders are provided between the tubes, so as to bring the entering steam at once into contact with a large percentage of the total tube area. A spiral partition or baffle is provided in order to make the steam take a spiral path and so to divide the surface that the area will be reduced in about the same proportion as the decrease in the quantity of steam to be condensed. The interior of this spiral baffle forms an air cooler, to cool the air and remove any water that may remain in it. The air is collected by this spiral baffle from over the full length of the steam space, and is removed at the two ends; therefore, there is no chance for the formation of air pockets at the ends of the condenser, as nearly always occurs in the old type. Such a collection of air would blanket a portion of the tube surface and prevent the steam from reaching it. Water-sealed drains are provided on the under side of the spiral baffle to remove any water that may be separated from the air.

The condenser is arranged for two passes of the circulating water. The cold water enters on the side opposite the steam inlet, and consequently a portion of it passes through the air-cooler tubes, thus giving the air as cold a temperature as possible. The partially heated water then passes through the tubes on the steam-inlet side of the condenser, thus bringing the hottest water into contact with entering steam. The water of condensation is removed from the bottom of the condenser.

The steam can enter either horizontally at the side, or vertically at the top. If the steam enters horizontally at the side, the water of condensation can be recovered, under normal load, at a temperature about one degree below the temperature of the vacuum; if at the top, about five degrees below. This is an important feature, since with old-design surface condensers, with top steam inlets, the water of condensation is recovered at a temperature from 15 degrees to 25 degrees below the vacuum temperature.

### HYDROFLO CENTRIFUGAL DRY-AIR PUMPS.

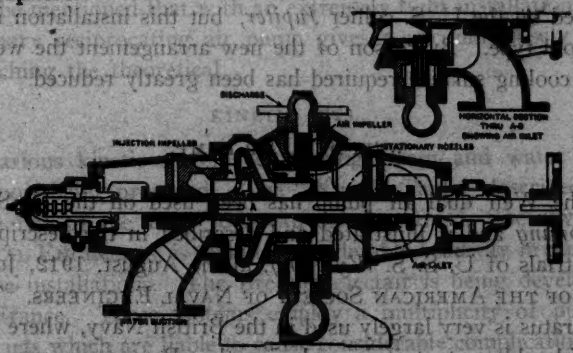
This pump is of the type in which the vacuum is produced by the flow of water, the air being entrapped between intermittent pistons of water moving at high velocity, which carry the air along with the water into the moving vanes of an impeller, where it is compressed, by centrifugal force, to the pressure of the atmosphere against which it is discharged. A small injection impeller lifts the water from the suction supply, and furnishes the initial velocity for entrapping the air. The smaller sizes are built with one water and one air impeller, this design being shown, Figs. 17, 18. The larger sizes are built

FIG. 17.

with two water impellers placed on either side of a double air impeller, the water and air being discharged through a common volute casing to the atmosphere. The double pump is arranged with two air and two water-suction openings.

The water is admitted through the suction opening and passes first through the injection impeller, which creates an initial pressure which is changed into velocity when the water is discharged through stationary nozzles into the air impeller. The air is admitted in between the stationary nozzles and the air impeller. The air impeller is of the multiple type, the vanes in each passage being shaped in a similar manner to

those of a standard centrifugal pump. The inner edges of the vanes during rotation cutoff plugs of water from the streams which are being projected into the impeller by the stationary nozzles above mentioned. After passing over the water nozzles, plugs of air are admitted to the impeller, then plugs of water, and so on. In this manner the passages in the air impeller are alternately filled with pistons of water and pistons of air. Both the water and the air are discharged by centrifugal force into the volute casing and from there into the atmosphere.




ALBERGER HYDROFLO CENTRIFUGAL DRY AIR PUMP
SECTIONAL VIEW

FIG. 18.

The direction of rotation of the air impeller is the same as that of a regular centrifugal pump, and the vanes so shaped as to disturb as little as possible the normal radial flow of the air and water; consequently, the efficiency of this impeller is comparable with the efficiency of a regular centrifugal pump handling water only. The whole operation is accomplished in one compact and complete unit. The direction of the flow of the water is not violently changed in an endeavor to project plugs of water through restricted passages or old-style ejectors at a velocity high enough to discharge same against the atmospheric pressure. Such a method of handling water, that is to say, to change violently its direction, is not economical in the use of power.

Due to the water impeller, which creates the initial velocity, the vacuum in the condenser can be completely lost and the *hydroflo* air pump will not lose its suction. This is an important feature, since, with other types of centrifugal air pumps, if the vacuum is once lost, it is necessary to completely re-prime the system before the vacuum can again be obtained. Means are provided for water sealing the stuffing boxes and the air impeller, to prevent air leakage into the pump while in operation.

An example of the Alberger condensing apparatus is installed on the U. S. collier *Jupiter*, but this installation is of the old type. By reason of the new arrangement the weight and cooling surface required has been greatly reduced.

### WEIR DUAL AIR PUMP.

The Weir dual air pump has been used on the U. S. S. *Wyoming* and is illustrated and described in the description and trials of U. S. S. *Wyoming*, in the August, 1912, JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS. This apparatus is very largely used in the British Navy, where to a considerable extent it is replacing the Parsons augmentor. The principal objection offered to the Parsons augmentor is that it is wasteful of steam and is somewhat heavy.

### OTHER LATE INSTALLATIONS.

The single-acting wet-air pump is used in connection with Parsons augmentors. Various arrangements for separating the air and water in separate chambers or cylinders of air pumps are also being employed. These systems often employ the general ideas made use of in the Weir dual air pump, namely, the air is taken by means of a separate suction from condenser or a branch from the suction and is handled by one cylinder of the air pump. Means for cooling the air either by passing it through a cooler similar to the Parsons augmentor condenser or by injecting cooled fresh water into the

dry-air-pump cylinder as employed in Weir's dual air pump are provided.

On destroyers built by the William Cramp & Sons Ship and Engine Building Company, a separate air suction is employed without the air cooler.

Of the two systems actually used for extended periods on our large naval vessels the dual air pump and the Parsons augmentor, there does not appear to be any decided advantage in either direction. Both systems are reliable and, with a properly designed condenser, give a satisfactory vacuum. It may also be mentioned that with an extremely tight installation the ordinary reciprocating air pump gives a vacuum closely approaching the theoretical.

### KINETIC JETS.

Various kinetic systems employing steam and water jets are now being proposed. The "Kinetic Tank," described on page 751 of the 1913 JOURNAL OF AMERICAN SOCIETY OF NAVAL ENGINEERS, is making considerable headway in power-house installations. The Breguet ejectair is being developed in France. These systems employ a multiplicity of pumps and jets which are liable to cause considerable complication on board ship, and it is therefore doubtful whether they will be extensively employed in marine work in the near future.

### MOST ACCEPTABLE SYSTEMS.

Of all the late systems the simplest are the Parsons augmentor and the Weir dual air pump, or its equivalent, as employed with pumps of other manufacturers. Both these systems have been in successful use in marine work and have been used in the British and American navies.

The Parsons augmentor is apparently more advantageous for small installations and for such service as on the dynamo condensers on men-of-war. The Parsons augmentor can also be readily added to an existing installation.

On large vessels, where the space and weight can be provided, it is probable that the systems which employ separate

dry-air pumps and which make careful provision for drawing the air from the condenser so that practically none is allowed to collect, may secure a somewhat better vacuum than that which is possible with the augmentor or dual air pump. The introduction of these compound, rotary or semi-rotary systems brings complications which are desirable to avoid if possible, while the practical gain is slight. The Westinghouse-Leblanc latest system will soon be tested in service on the *Neptune*. The Scout Cruiser *Salem* has had a semi-rotary dry-vacuum system which has been generally satisfactory and which was installed in 1907.

### POSITION OF CONDENSER IN REFERENCE TO ENGINE.

It would appear desirable to have the condenser in a position below that of the exhaust from the engine, provided the necessary drop from bottom of condenser to air-pump suction can be secured. The drop from the bottom of condenser to air-pump suction is, however, more important than the drop from the exhaust connection to the condenser.

A condenser should be as close as possible to the exhaust. A long exhaust pipe produces undesirable friction and increases back pressure. A low position of condenser also permits of better drainage of cylinders, and in cases where the condenser must be placed well above the engine a drain connection should be supplied from the bottom of exhaust pocket to the air-pump suction. This drain could have a special cooler fitted to it or it could run through the augmentor condenser.

### WATER AGITATING DEVICES.

The question of using whirling plates or corrugated tubes for condensers has sometimes been advanced, though as far as is known they have not been tried for marine work. There are well founded objections on the ground that both these devices might cause the tubes to become clogged and thus create conditions much worse than any advantage that might possibly be gained.

### COOLING SURFACE—CLEANLINESS OF.

The efficiency of the condenser is very largely affected by the cleanliness of the condensing surface, both the steam side and the water side.

*Steam side.*—Where oil is used for internal lubrication it will be carried over with the steam and a considerable part of it will be deposited on the surface of the condenser tubes. The presence of this oil will render the cooling surface much less efficient. Therefore where there is this accumulation of oil deposit the condenser should be periodically boiled out. This is done by adding a quantity of soda and boiling the water in the condenser by means of a steam connection. The soda will dissolve the oil on the tubes and hold it in solution. After being boiled out the mixture of oil and water should be drained off.

*Water side.*—The water side of the condenser may collect considerable sand, ashes, mud and marine growth and, of course, any considerable amount of this will detract from the cooling ability of the surface. The inside of the tubes may be cleaned by running brushes through them. An easier way is to clean them with an air blast should compressed air be available. A thorough cleaning of the surface will add a great deal to the efficiency of the condenser.

### PROTECTING TUBES FROM STEAM BLAST.

If the steam from any condenser connection is allowed to impinge directly on the tubes it is very likely to cut and erode the tubes. For this reason all connections, such as the auxiliary exhaust, discharges from traps, etc., should be fitted with a protecting baffle plate that will protect the tubes from the direct blast of the steam.

### TESTING CONDENSERS FOR LEAKS.

The two most usual methods of testing a condenser for tightness are by candle with vacuum on condenser and by water pressure.

The first mentioned is the easier and quicker, but it is not

as reliable. The examination plates over the tubes are removed and the condenser drained on water side. The air pump is started and a vacuum of several inches of mercury secured. Then a lighted candle is held over the tube ends, stay connections and any other place where there might be a leak. A leak will cause air to be taken in, and this will also be indicated by the flame being drawn in. Any considerable leak will be indicated in this way, but a very small leak cannot be well located by this method.

*By water pressure.*—Here the heads or examination plates are removed and then the steam side filled with water as high as possible without putting it in the engine cylinders. Any leak will be indicated by water trickling out.

*By air pressure.*—In turbine installations, where the turbine exhaust frequently discharges into the side of the condenser, the water-pressure method cannot be used without special arrangements for blanking off the exhaust. In such installations an air pressure can be used. Steam is put in the turbine glands to avoid air leakage and then condenser and turbine casing are put under an air pressure of several pounds per square inch. While under this pressure the tube ends, joints, etc., can be examined for any air leakage. A candle light may be used to indicate any leakage, since even a very slight issue of air will cause the flame to flicker. Another means for definitely indicating air leaks is to cover suspected places with soapsuds. Any slight issue of air will indicate itself at once by its action on the soap films. A very slight leak can readily be indicated by the above method, which is clean and convenient.

### CORROSIVE ACTION.

To guard against corrosive action of sea water zinc plates are usually fitted in the water spaces of the condenser heads. These apparently arrest corrosion, but they do not entirely prevent it. Where the heads are of cast iron there is usually very little corrosion on the tubes and tube sheets. This fact has led to the idea of preventing corrosion by fitting cast-iron pieces instead of zincs in these spaces.

When cast iron is used for condenser shell and heads it is made extra thick to allow for such amount of corrosion as may take place.

### SALT LEAKS.

A most essential matter in marine-condenser design and construction is the precaution taken to guard against salt leaks.

A leaky condenser is a thing to be shunned and will counterbalance any advantage otherwise gained. The chief sources of salt leakage are the condenser-stay connections and the tube ends. The number of stays required can be greatly reduced by careful design, and it is contended that marine condensers should be built without any of the stays passing entirely through the tube sheet. It has further been demonstrated that if the tube sheets are stayed to the heads and the tubes expanded into the tube sheets, stays in the vapor space will be entirely unnecessary—the tubes will give ample stiffness. If this is done salt leakage is eliminated, except that due to holes or splits in the tubes.

Condensers have usually been fitted with packed tubes at both ends. The tube packing is always a source of some leakage. To reduce this many recently constructed condensers have a gland at one end only.

Expanded tube condensers where the tubes are bent to the arc of a large circle have been used and are entirely successful. In some condensers tubes have been threaded and screwed into the tube sheet. This latter is not a successful practice since there is a very great likelihood of tubes corroding at the threads and of leaking in the threads.

In a stayless condenser with expanded tubes the chances of any salt leaks are a minimum. Such condensers can and are being built at no large extra expense, but few are making use of their advantages. Such condensers have been used on a number of merchant vessels and all destroyers built by the Bath Iron Works for the past five years, and they have been preëminently successful and salt leaks are practically eliminated on these vessels. It is now the practice to call for expanded tube condensers on U. S. destroyers.

# EMERGENCY REPAIR TO SUBMARINES.

By Lieutenant (J. G.) C. N. Hinkamp, Member.

## REPAIRS TO SPIRAL GEARS.

While en route to Charleston, S. C., on the U. S. S. *Octopus* in December, 1909, the port engine suddenly stopped. A hasty examination showed that the cam shaft did not revolve when the engine was jacked over. This was caused by the teeth on the driven spiral gear having stripped. Immediate repairs were necessary owing to the state of the battery charge, inability to charge battery with the starboard engine on account of the condition of the motor, and the necessity for keeping the starboard engine on the propeller because of the weather conditions, which were bad. There was no spare gear on board for this engine. The stripped gear was removed and carefully examined. Three teeth were gone completely and two adjacent ones in bad condition. The gear was filed off smoothly and the location of the teeth scribed on the wheel. By rigging a drill press with a breast drill, a lead line and a stanchion, holes were drilled in the gear as per sketch "A," and tapped. The tap was a "starting tap," and the only one of that size on board. All the holes were "started," then the tap was broken off and used as a "bottoming" tap. An iron rod used for sounding fuel tanks was then threaded, screwed in the holes and sawed off at the height of the tooth. This process was repeated until all the holes were plugged. The projecting parts were then filed to the shape of the teeth as nearly as was possible, the other teeth faired up, the gear was reinstalled and, after setting the cam shaft, the engine was started. This engine ran successfully while charging battery and later on the propeller until the

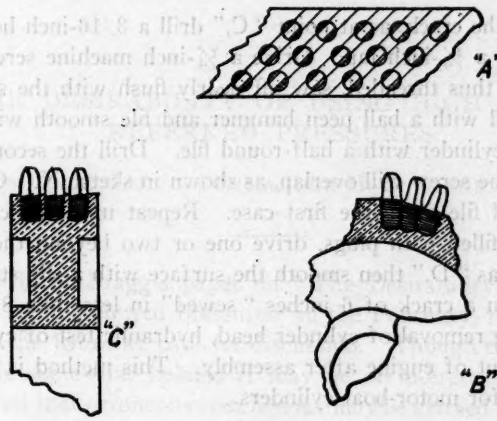



FIG. 1.

Navy Yard was reached the next day. The time for repairs, from stopping of engine to starting up, was less than 2 hours.

The accompanying sketches, Fig 1, indicate the methods used.

## CYLINDER REPAIRS.

The following method of making temporary repairs to cracked gasoline-engine cylinders, where the crack has not been too bad, has been used with success on the submarines now at Panama. The process is known as "sewing." Suppose the crack to be "AB" as shown on Fig. 2. Somewhere

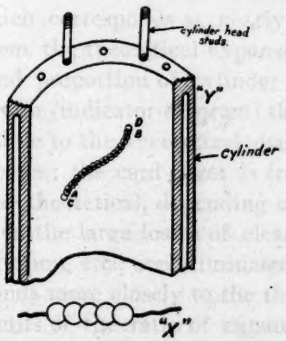
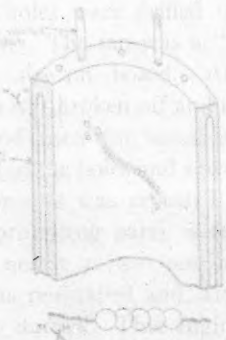



FIG. 2.

beyond the crack, as at point "C," drill a 3/16-inch hole and tap with a ¼-inch tap. Drive a ¼-inch machine screw into the hole thus threaded, saw off nearly flush with the surface, peen well with a ball peen hammer and file smooth with surface of cylinder with a half-round file. Drill the second hole so that the screw will overlap, as shown in sketch X. Cut off, peen and file as in the first case. Repeat until the crack is entirely filled with plugs, drive one or two beyond the crack to point as "D," then smooth the surface with a slip stone. I have seen a crack of 6 inches "sewed" in less than 8 hours, including removal of cylinder head, hydraulic test of cylinder, and tryout of engine after assembly. This method is recommended for motor-boat cylinders.



# THE DESIRABILITY OF USING HIGH MEAN REFERRED PRESSURES.

DISCUSSION BY LIEUT. COMMANDER H. C. DINGER, U. S. N.

Mr. E. M. Bragg's paper on "The Desirability of Using High Mean Referred Pressures" in the February, 1913, number of the JOURNAL calls for discussion. Though this discussion is somewhat belated it may be of interest, since it is believed that erroneous conclusions may be arrived at if some of the matters presented were to go unquestioned. A close examination into the data presented and other data equally well at hand fails to show that it is economical to have a high mean referred pressure by reducing the ratio of expansion.

To begin with, the mean referred pressure is really a result, not a premise. The best or most economical referred pressure will vary as the conditions under which the steam is used vary.

The most efficient mean referred pressure, or rather the mean referred pressure that corresponds to the most efficient condition within the limit of initial and final temperature, is that pressure which corresponds as nearly as possible to the work diagram from the theoretical expansion curve.

Good design and proportion of cylinder aims to secure an actual work diagram (indicator diagram) that will correspond as closely as possible to the theoretical diagram. There are always certain losses; the card effect is from 50 per cent. to 75 per cent. of the theoretical, depending on the efficiency of the design. When the large losses of clearances, initial condensation, port friction, etc., are eliminated the actual work diagram corresponds more closely to the theoretical diagram.

Within the limits of the ratio of expansion of any given engine a high mean referred pressure does correspond to econ-

omy. But this holds only for the same number of expansions. As the number of expansions is increased the referred pressure is decreased, both that theoretically possible and that practically attainable.

In an engine where the cut-off and the ratio of cylinders is such as to give eight expansions, and with the same initial and final temperature, the higher the mean referred pressure the better the economy. That is, the mean referred pressure is then a scale showing relative efficiency for that engine and particular ratio of expansion. It is then a function of the card effect.

If the cylinder ratio is increased so as to increase the expansions the theoretical mean referred pressure is likewise changed, having been lowered on account of increased expansion.

Other things being the same, an increase in the number of expansions will increase the economy and lower the mean referred pressure. There is absolutely no way of getting away from this; it is as definite as the law of gravity. But, if other things are not the same, a lower mean referred pressure may sometimes mean less economy. If initial condensation losses, clearance losses, internal and port friction losses are increased at the same time that the number of expansions are increased, it may well be possible that a certain engine with an increased number of expansions and a lower mean referred pressure will be less economical than another with a less number of expansions and higher referred pressure. In any case, the mean referred pressure is an incident and a consequence and is not a basis. It is raised or lowered in any given engine by the state of efficiency of its various parts. A leaky piston will lower it; a tight piston raise it. Moist steam will lower it; dry or superheated steam will raise it.

Mr. Bragg cites the performance of the *Birmingham* and the *Delaware* to prove that a high mean referred pressure and low ratio of expansion is economical. This comparison really proves nothing of the kind. All that it does prove is that the *Delaware's* engine is about 20 per cent. more eco-

nomical than the *Birmingham's* engine. The other differences in condition, for instance, the higher piston speed, much shorter stroke, almost double-cylinder clearances, less efficient valves, and pistons having greater leakage, to say nothing of the fact that the *Birmingham* was using moist steam at 233 pounds while the *Delaware* was using steam superheated to about 80 degrees F. with an initial pressure of 263 pounds, and the fact that the *Birmingham's* engines weigh just about one-half of the weight of the *Delaware's* engines. The differences in condition of the two engines are so many that a comparison of water consumption with a view of proving that it is due to one point of difference is futile. Most of the difference is due to the use of superheat greatly reduced clearances is another material factor. In fact, these two items account for at least three-fourths of the difference in steam consumption.

But suppose we do take a similar engine that has a larger ratio of expansion and lower mean referred pressure, and what do we get? Let us take the *Michigan's* engine. This is an engine working under similar conditions, same quality of steam, same stroke, clearance, valve design, etc., and what do we find? We find the *Michigan* decidedly more economical than the *Delaware.*

Take the *Delaware* on her 19-knot trial when the mean referred pressure was lower, and we find with the lower referred pressure there is better economy. Here are the results:

| | * Coal per I.H.P. | Ratio of expansion. | M.E.P. |
|---|---|---|---|
| *Delaware*, . . . | 1.88 | 9.1 | 51 |
| *Michigan*, . . . | 1.51 | 12.9 | 35 |
| *Delaware*, 19 knots, . | 1.79 | 9.1 | 34 |
| *North Carolina*, . . | 2.74 | 8.3 | 53.2 |
| *Birmingham*, . . | 2.28 | 11.3 | 34.8 |
| *Texas*, . . . . | 1.73 | 11.4 | 43.59 |

* Water consumption cannot be compared as it was not measured on the *Michigan* and *North Carolina*. The boilers of the vessels other than *Birmingham* are, however, identical, except no superheat on *North Carolina*, and under these conditions the coal consumption ought to be practically as good a comparison.

Or let us take the engines of the *North Carolina*, same stroke, piston speed and steam pressure as the *Delaware*, but with a ratio of expansion slightly less than that of the *Delaware*.

The mean referred pressure is even greater than that of the *Delaware*, but the economy is nowhere near as good.

The relative economy of the *Michigan* and the *North Carolina*, whose engines are similar, except for use of superheat, clearances and proportion of cylinders, show a decided proof that increased ratio of expansion and consequent lower mean effective pressures enhance economy within the limits there used.

The efficiency of any steam engine cylinder is indicated by the card effect. If, by the use of a very large ratio, the card effect is greatly decreased, a point is no doubt reached where further increase in expansions produces loss. For single cylinders this has been experimentally determined to be between three and four expansions. For multiple-cylinder engines, using 265 pounds pressure, it is at least 15; with a mean effective pressure of about 36 pounds, vacuum 28 inches.

Sir Charles Parsons, in an address before the Northeast Coast Shipbuilders, stated that he considered the probable limit to about 16 expansions.

The most economical reciprocating engines on any of our large battleships are those of the *Michigan* and *South Carolina*, and they employ a large cylinder ratio and correspondingly low mean referred pressure.

In connection with this matter it may be of interest to refer to certain tables prepared some years ago by Mr. A. H. Raynal, of the Bureau of Steam Engineering. These tables give the *mean effective pressures*, *expansions* and *steam consumption per horsepower* for various initial and terminal pressures, and different percentages of card effect, as applied to marine reciprocating engines using saturated steam.

The steam consumptions actually found on engines correspond very closely to these figures in all cases. An examination of these tables clearly shows that with the same card

effect the economy increases as the ratio of expansion is increased and as the M.E.P. is lowered.

In practice there are many elements of design that modify the card effect actually secured; jackets, superheat, clearance and the efficiency of the valve gear all have their influence.

A very high ratio of expansion will no doubt introduce some conditions which are undesirable, but for an initial pressure of 250 pounds a ratio of expansion of 15 and somewhat higher may be used without arriving at the point where expansion is carried too far. While it was at one time demonstrated that expansion in a single cylinder should not be carried beyond 4, when the engine is compounded this is changed, and increased expansions can be used to advantage, as they have been in the most economical reciprocating engines that have been built.

Mean referred pressure is, however, an indefinite thing, and, as said before, is a consequence of the design and not a basis. The ratio of expansion having been selected, the *mean referred pressure* follows as a consequence, and of course the higher it can be made for any determined ratio of expansion the better the card effect and the better the economy. It is, of course, desirable to increase the card effect and also the M.E.P. if the same quantity of steam is being supplied. But when the M.E.P. is increased by using a lesser number of expansions and a larger amount of steam, increased economy does not necessarily follow.

The engines of the *Birmingham* are cited, and the statement is made, "that had the *Birmingham's* engines been designed to develop maximum power with 8 expansions and a referred pressure of about 55 pounds, their steam consumption would have been reduced 20 per cent." From all data that is available the contrary would have been the result. The referred pressure could have been increased by putting more steam through the engine, which was possible and was actually done in service. But this would not improve the economy. It has been repeatedly demonstrated that marine reciprocating engines are considerably more economical at

from 80 to 90 per cent. of their maximum power than at maximum power.

If instead of advocating use of higher M.R.P. the argument had been advanced for securing better card effects the soundness of the argument could not be questioned. If, on the other hand, it can be demonstrated that increased expansions lower the card effect to a greater degree than the gain that can be secured by the greater expansion of steam it will then be shown that the profitable limit of expansions has been reached. Sufficient experiments with large marine engines have not been made to indicate where this point is, but the most economical engines known of employ a higher ratio of expansion than that recommended by Mr. Bragg. The *Michigan* has a more economical engine than the *Delaware* and a considerable higher ratio of expansion is employed. The *Texas* engine is more economical than the *Delaware*, and this also employs a larger ratio of expansion. It will be seen, when engines of similar stroke and construction are compared, that the highest ratio of expansion gives the most economy. As far as is known the *Michigan* and *South Carolina* have the highest ratio of any late triple-expansion marine engines that have been built. And they are the most economical marine triple that we have in the Navy, or anywhere else as far as is known.

On the full-power trials in service, in which each vessel developed the greatest power to date, the coal for all purposes per I.H.P. of main engines was as follows: *Delaware*, 1.80; *Michigan*, 1.48. These results are somewhat better than those on the official trials, but the increased economy of the *Michigan's* engines is maintained in about the same proportion. If we assume that boiler conditions are equal the *Michigan's* engines give a water consumption of 11 pounds per I.H.P. per hour. This, of course, is an estimate, and it is doubtful whether the *Michigan's* engine is actually quite as economical as this. But the consistent superiority in economy over the engines of the *Delaware* indicates that the water consumption at full power does not materially exceed 12 pounds per I.H.P.

## Expansions, M.E.P. and Steam Consumption per H.P. per Hour for 100 Per Cent. Card Effect.

| Terminal pressure. | Steam pressure. | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | Above atmos. | 50 | 75 | 100 | 125 | 150 | 175 | 200 | 225 | 250 |
| | Absolute. | 65 | 90 | 115 | 140 | 165 | 190 | 215 | 240 | 265 |
| 5 | Expansion... | 13 | 18 | 23 | 28 | 33 | 38 | 43 | 48 | 53 |
| | M.E.P..... | 17.82 | 19.45 | 20.67 | 21.66 | 22.48 | 23.19 | 23.81 | 24.36 | 24 85 |
| | Steam ..... | 10.59 | 9.71 | 9.13 | 8.72 | 8.4 | 8.14 | 7.93 | 7.75 | 7.6 |
| 6 | Expansion... | 10.83 | 15 | 19.17 | 23.33 | 27.5 | 31.67 | 35.83 | 40 | 44.17 |
| | M.E.P..... | 20.29 | 22.25 | 23.71 | 24.9 | 25.88 | 26.73 | 27.47 | 28.13 | 28.73 |
| | Steam ..... | 11.05 | 10.08 | 9.46 | 9.01 | 8.66 | 8.39 | 8.17 | 7.98 | 7.81 |
| 7 | Expansion... | 9.29 | 12.86 | 16.43 | 20 | 23.57 | 27.14 | 30.71 | 34.29 | 37 86 |
| | M.E.P..... | 22.59 | 24.87 | 26.58 | 27.97 | 29.12 | 30.11 | 30.97 | 31.74 | 32.44 |
| | Steam ..... | 11.48 | 10.44 | 9.76 | 9.28 | 8.91 | 8.62 | 8.38 | 8.18 | 8 |
| 8 | Expansion... | 8.13 | 11.25 | 14.38 | 17.5 | 20.63 | 23.75 | 26 88 | 30 | 33.13 |
| | M.E.P..... | 24.76 | 27.36 | 29.32 | 30.92 | 32.21 | 33.34 | 34.33 | 35.21 | 36 |
| | Steam ..... | 11 88 | 10.76 | 10.04 | 9 52 | 9.14 | 8.83 | 8.57 | 8.36 | 8.17 |
| 9 | Expansion... | 7.22 | 10 | 12.78 | 15.56 | 18.33 | 21.11 | 23.89 | 26.67 | 29.44 |
| | M.E.P..... | 26.79 | 29.72 | 31.92 | 33.7 | 35.18 | 36.45 | 37.56 | 38.5 | 39.44 |
| | Steam ..... | 12.27 | 11.06 | 10.3 | 9.76 | 9.35 | 9.02 | 8.75 | 8.54 | 8.34 |
| 10 | Expansion... | 6.5 | 9 | 11.5 | 14 | 16.5 | 19 | 21.5 | 24 | 26.5 |
| | M.E.P..... | 28.72 | 31.97 | 34.41 | 36.39 | 38.03 | 39 44 | 40.68 | 41.78 | 42.77 |
| | Steam ..... | 12.65 | 11.36 | 10.55 | 9 98 | 9.55 | 9 21 | 8.93 | 8.69 | 8.49 |
| 11 | Expansion... | 5.91 | 8.18 | 10.45 | 12.73 | 15 | 17.27 | 19.55 | 21.82 | 24.1 |
| | M.E.P..... | 30.54 | 34.12 | 36.81 | 38.99 | 40.8 | 42.35 | 43.7 | 44.92 | 46.01 |
| | Steam ..... | 13.01 | 11.64 | 10.79 | 10.19 | 9.74 | 3.38 | 9.09 | 8.84 | 8 63 |
| 12 | Expansion... | 5.42 | 7.5 | 9.58 | 11.67 | 13.75 | 15.83 | 17.92 | 20 | 22.08 |
| | M.E.P..... | 32.27 | 36.18 | 39.1 | 41.48 | 43.45 | 45.14 | 46 63 | 47.95 | 49.14 |
| | Steam ..... | 13.36 | 11.92 | 11.03 | 10.4 | 9.92 | 9.55 | 9.25 | 8.99 | 8.78 |
| 13 | Expansion... | 5 | 6.92 | 8.85 | 10.77 | 12.69 | 14.62 | 16.54 | 18.46 | 20.38 |
| | M.E.P..... | 33.92 | 38.15 | 41.32 | 43 9 | 46.03 | 47.87 | 49.48 | 50 91 | 52.19 |
| | Steam ..... | 13.7 | 12.19 | 11.25 | 10.59 | 10.1 | 9.7 | 9.4 | 9.13 | 8.91 |
| 14 | Expansion... | 4.64 | 6.43 | 8.21 | 10 | 11.79 | 13.57 | 15.36 | 17.14 | 18.93 |
| | M.E.P..... | 35.49 | 40.04 | 43.47 | 46.24 | 48.51 | 50.48 | 52.24 | 53.74 | 55.17 |
| | Steam ..... | 14.04 | 12.45 | 11.46 | 10.78 | 10.28 | 9.87 | 9.54 | 9.28 | 9.03 |
| 15 | Expansion... | 4.33 | 6 | 7.67 | 9.33 | 11 | 12.67 | 14.33 | 16 | 17.67 |
| | M.E.P..... | 36.99 | 41.88 | 45.53 | 48.5 | 50.97 | 53 08 | 54.94 | 56.59 | 58.08 |
| | Steam ..... | 14.38 | 12.7 | 11.68 | 10.97 | 10.43 | 10.02 | 9.68 | 9.4 | 9.16 |
| 16 | Expansion... | 4.06 | 5.63 | 7.19 | 8.75 | 10.31 | 11.88 | 13.44 | 15 | 16.56 |
| | M.E.P..... | 38.43 | 43.64 | 47.54 | 50.71 | 53.33 | 55.59 | 57.57 | 59.33 | 60.91 |
| | Steam ..... | 14.71 | 12.95 | 11.79 | 11.15 | 10.6 | 10.17 | 9.82 | 9.53 | 9.28 |
| 17 | Expansion... | 3.82 | 5.29 | 6.76 | 8 24 | 9.71 | 11.18 | 12.64 | 14.12 | 15.59 |
| | M.E.P..... | 39.8 | 45.33 | 49.48 | 52.87 | 55.64 | 58.04 | 60.14 | 62.01 | 63.69 |
| | Steam ..... | 15.04 | 13.2 | 12.09 | 11 32 | 10.76 | 10.31 | 9.95 | 9.65 | 9.4 |
| 18 | Expansion... | 3.61 | 5 | 6.39 | 7.78 | 9.17 | 10.56 | 11.94 | 13.33 | 14.72 |
| | M.E.P..... | 41.11 | 46.97 | 51.36 | 54.92 | 57.88 | 60.42 | 62.64 | 64 66 | 66.08 |
| | Steam ..... | 15.36 | 13.44 | 12.29 | 11.5 | 10 91 | 10.45 | 10.08 | 9.76 | 9.51 |
| 19 | Expansion... | 3.42 | 4.73 | 6.05 | 7 37 | 8.68 | 10 | 11.32 | 12 63 | 13.95 |
| | M.E.P..... | 42.37 | 48.55 | 53.19 | 56.94 | 60.07 | 62 75 | 65.1 | 67.19 | 69.07 |
| | Steam ..... | 15.68 | 13.68 | 12.48 | 11.67 | 11.06 | 10.59 | 10.21 | 9.89 | 9 62 |
| 20 | Expansion... | 3.25 | 4 5 | 5.75 | 7 | 8.25 | 9.5 | 10.75 | 12 | 13.25 |
| | M.E.P..... | 43.57 | 50 08 | 54.96 | 58.92 | 62.2 | 65.03 | 67.5 | 69.7 | 71.68 |
| | Steam ..... | 16 | 13.92 | 12.68 | 11.84 | 11.21 | 10.72 | 10.33 | 10 | 9.72 |
| 21 | Expansion... | 3.09 | 4.29 | 5.48 | 6.67 | 7.86 | 9.05 | 10.24 | 11.42 | 12.62 |
| | M.E.P..... | 44.73 | 51.56 | 56.69 | 60.84 | 64.29 | 67.25 | 69.85 | 72.16 | 74.24 |
| | Steam ..... | 16.32 | 14 16 | 12.89 | 12 | 11.35 | 10.89 | 10.45 | 10.12 | 9.83 |
| 22 | Expansion... | 2.95 | 4.1 | 5.23 | 6.36 | 7.5 | 8.64 | 9.77 | 10.91 | 12.05 |
| | M.E.P..... | 45.83 | 52 99 | 58.36 | 62.71 | 66.33 | 69.43 | 72.15 | 74.57 | 76.75 |
| | Steam ..... | 16.64 | 14.39 | 13.06 | 12.16 | 11.5 | 10 98 | 10.57 | 10.22 | 9.93 |
| 23 | Expansion... | 2.83 | 3.19 | 5 | 6.09 | 7.17 | 8.26 | 9.35 | 10.43 | 11.52 |
| | M.E.P..... | 46.91 | 54.39 | 60 | 64.55 | 68.33 | 71.57 | 74.41 | 76.94 | 79.22 |
| | Steam ..... | 16.95 | 14.62 | 13.25 | 12.32 | 11.64 | 11.11 | 10 69 | 10.33 | 10.04 |
| 24 | Expansion... | 2.71 | 3.75 | 4.79 | 5.83 | 6.46 | 7 92 | 8.54 | 10 | 11.04 |
| | M.E.P..... | 47.91 | 55.72 | 61.57 | 66.33 | 70.27 | 73.66 | 76.62 | 79.26 | 81.64 |
| | Steam ..... | 17.27 | 14.85 | 13.44 | 12 47 | 11.78 | 11.28 | 10 8 | 10.44 | 10.14 |
| 25 | Expansion... | 2 6 | 3.6 | 4.6 | 5.6 | 6.6 | 7.6 | 8.6 | 9.6 | 10.6 |
| | M.E.P..... | 48.89 | 57.03 | 63.12 | 68.07 | 72.18 | 75.7 | 78 79 | 81.55 | 84.02 |
| | Steam ..... | 17.59 | 15.08 | 13.62 | 12 63 | 11.91 | 11.35 | 10 91 | 10.54 | 10.23 |
| 26 | Expansion... | 2.5 | 3.46 | 4.42 | 5.38 | 6 35 | 7.31 | 8.27 | 9.23 | 10.19 |
| | M.E.P..... | 49.82 | 58.28 | 64.62 | 69.77 | 74.11 | 77.71 | 80.92 | 83.79 | 86.36 |
| | Steam ..... | 17.9 | 15.27 | 13.8 | 12.79 | 12.03 | 11.48 | 11.02 | 10.65 | 10.33 |
| 27 | Expansion... | 2.47 | 3 33 | 4.26 | 5.18 | 6.11 | 7.04 | 7.96 | 8.89 | 9.81 |
| | M.E.P..... | 50.65 | 59 45 | 66.05 | 71.41 | 75 85 | 79.67 | 83.01 | 85.99 | 88.66 |
| | Steam ..... | 18 25 | 15.55 | 13.99 | 12.94 | 12.15 | 11.6 | 11.13 | 10.75 | 10.42 |
| 28 | Expansion... | 2.32 | 3.21 | 4.11 | 5 | 5.89 | 6 79 | 7.68 | 8.57 | 9.46 |
| | M.E.P..... | 51.58 | 60.69 | 67.52 | 73.01 | 77 66 | 81.62 | 85.08 | 88.16 | 90 93 |
| | Steam ..... | 18.54 | 15.76 | 14.16 | 13.1 | 12.31 | 11.72 | 11.24 | 10.85 | 10.52 |
| 29 | Expansion... | 2.24 | 3.1 | 3.97 | 4.83 | 5.69 | 6.55 | 7.41 | 8.28 | 9.14 |
| | M.E.P..... | 52.4 | 61.85 | 68.92 | 74.66 | 79.42 | 83.51 | 87.1 | 90.29 | 93.16 |
| | Steam ..... | 18.86 | 15.98 | 14.34 | 13.24 | 12.45 | 11.84 | 11.35 | 10.95 | 10.61 |
| 30 | Expansion... | 2.17 | 3 | 3.83 | 4.67 | 5.5 | 6.33 | 7.17 | 8 | 8.83 |
| | M.E.P..... | 53.2 | 62.96 | 70.28 | 76.22 | 81.15 | 85.38 | 89.1 | 92.39 | 95.36 |
| | Steam ..... | 19.18 | 16.2 | 14.51 | 13.39 | 12.57 | 11.95 | 11.45 | 11.04 | 10.7 |

from 80 to 90 per cent. of their maximum power than at maximum power.

If instead of advocating use of higher M.R.P. the argument had been advanced for securing better card effects the soundness of the argument could not be questioned. If, on the other hand, it can be demonstrated that increased expansions lower the card effect to a greater degree than the gain that can be secured by the greater expansion of steam it will then be shown that the profitable limit of expansions has been reached. Sufficient experiments with large marine engines have not been made to indicate where this point is, but the most economical engines known of employ a higher ratio of expansion than that recommended by Mr. Bragg. The *Michigan* has a more economical engine than the *Delaware* and a considerable higher ratio of expansion is employed. The *Texas* engine is more economical than the *Delaware*, and this also employs a larger ratio of expansion. It will be seen, when engines of similar stroke and construction are compared, that the highest ratio of expansion gives the most economy. As far as is known the *Michigan* and *South Carolina* have the highest ratio of any late triple-expansion marine engines that have been built. And they are the most economical marine triple that we have in the Navy, or anywhere else as far as is known.

On the full-power trials in service, in which each vessel developed the greatest power to date, the coal for all purposes per I.H.P. of main engines was as follows: *Delaware*, 1.80; *Michigan*, 1.48. These results are somewhat better than those on the official trials, but the increased economy of the *Michigan's* engines is maintained in about the same proportion. If we assume that boiler conditions are equal the *Michigan's* engines give a water consumption of 11 pounds per I.H.P. per hour. This, of course, is an estimate, and it is doubtful whether the *Michigan's* engine is actually quite as economical as this. But the consistent superiority in economy over the engines of the *Delaware* indicates that the water consumption at full power does not materially exceed 12 pounds per I.H.P.

## Expansions, M.E.P. and Steam Consumption per H.P. per Hour for 100 Per Cent. Card Effect.

| Terminal pressure. | Steam pressure. | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | Above atmos. | 50 | 75 | 100 | 125 | 150 | 175 | 200 | 225 | 250 |
| | Absolute. | 65 | 90 | 115 | 140 | 165 | 190 | 215 | 240 | 265 |
| 5 | Expansion... | 13 | 18 | 23 | 28 | 33 | 38 | 43 | 48 | 53 |
| | M.E.P..... | 17.82 | 19.45 | 20.67 | 21.66 | 22.48 | 23.19 | 23.81 | 24.36 | 24 85 |
| | Steam ..... | 10.59 | 9.71 | 9.13 | 8.72 | 8.4 | 8.14 | 7.93 | 7.75 | 7.6 |
| 6 | Expansion... | 10.83 | 15 | 19.17 | 23.33 | 27.5 | 31.67 | 35.83 | 40 | 44.17 |
| | M.E.P..... | 20.29 | 22.25 | 23.71 | 24.9 | 25.88 | 26.73 | 27.47 | 28.13 | 28.73 |
| | Steam ..... | 11.05 | 10.08 | 9.46 | 9.01 | 8.66 | 8.39 | 8.17 | 7.98 | 7.81 |
| 7 | Expansion... | 9.29 | 12.86 | 16.43 | 20 | 23.57 | 27.14 | 30.71 | 34.29 | 37 86 |
| | M.E P..... | 22.59 | 24.87 | 26.58 | 27.97 | 29.12 | 30.11 | 30.97 | 31.74 | 32.44 |
| | Steam . .. .... | 11.48 | 10.44 | 9.76 | 9.28 | 8.91 | 8.62 | 8 38 | 8.18 | 8 |
| 8 | Expansion... | 8.12 | 11.25 | 14.38 | 17.5 | 20.63 | 23.75 | 26 88 | 30 | 33.13 |
| | M.E.P..... | 24.76 | 27.36 | 29.32 | 30.92 | 32.21 | 33.34 | 34.33 | 35.21 | 36 |
| | Steam ..... | 11 88 | 10.76 | 10.04 | 9 52 | 9.14 | 8.83 | 8.57 | 8.36 | 8.17 |
| 9 | Expansion... | 7.22 | 10 | 12.78 | 15.56 | 18.33 | 21.11 | 23.89 | 26.67 | 29.44 |
| | M.E.P..... | 26.79 | 29.72 | 31.92 | 33.7 | 35.18 | 36.45 | 37.56 | 38.5 | 39.44 |
| | Steam ..... | 12.27 | 11.06 | 10.3 | 9.76 | 9.35 | 9.02 | 8.75 | 8.54 | 8.34 |
| 10 | Expansion... | 6.5 | 9 | 11.5 | 14 | 16.5 | 19 | 21.5 | 24 | 26.5 |
| | M.E.P..... | 28.72 | 31.97 | 34.41 | 36.39 | 38.03 | 39 44 | 40.68 | 41.78 | 42.77 |
| | Steam ..... | 12.65 | 11.36 | 10.55 | 9 98 | 9.55 | 9 21 | 8.93 | 8.69 | 8.49 |
| 11 | Expansion... | 5.91 | 8.18 | 10.45 | 12.73 | 15 | 17.27 | 19.55 | 21.82 | 24.1 |
| | M.E.P..... | 30.54 | 34.12 | 36.81 | 38.99 | 40.8 | 42.35 | 43.7 | 44.92 | 46.01 |
| | Steam ..... | 13.01 | 11.64 | 10.79 | 10.19 | 9.74 | 3.38 | 9.09 | 8.84 | 8 63 |
| 12 | Expansion... | 5.42 | 7.5 | 9.58 | 11.67 | 13.75 | 15.83 | 17.92 | 20 | 22.08 |
| | M.E.P..... | 32.27 | 36.18 | 39.1 | 41.48 | 43.45 | 45.14 | 46 63 | 47.95 | 49.14 |
| | Steam ..... | 13.36 | 11.92 | 11.03 | 10.4 | 9.92 | 9.55 | 9.25 | 8.99 | 8.78 |
| 13 | Expansion... | 5 | 6.92 | 8.85 | 10.77 | 12.69 | 14.62 | 16.54 | 18.46 | 20.38 |
| | M.E.P..... | 33.92 | 38.15 | 41.32 | 43 9 | 46.03 | 47.87 | 49.48 | 50 91 | 52.19 |
| | Steam ..... | 13.7 | 12.19 | 11.25 | 10.59 | 10.1 | 9.7 | 9.4 | 9.13 | 8.91 |
| 14 | Expansion... | 4.64 | 6.43 | 8.21 | 10 | 11.79 | 13.57 | 15.36 | 17.14 | 18.93 |
| | M.E.P..... | 35.49 | 40.04 | 43.47 | 46.24 | 48.51 | 50.48 | 52.24 | 53 74 | 55.17 |
| | Steam ..... | 14.04 | 12.45 | 11.46 | 10.78 | 10.28 | 9.87 | 9.54 | 9.28 | 9.03 |
| 15 | Expansion... | 4.33 | 6 | 7.67 | 9.33 | 11 | 12.67 | 14.33 | 16 | 17.67 |
| | M.E.P..... | 36.99 | 41.88 | 45.53 | 48.5 | 50.97 | 53 08 | 54.94 | 56.59 | 58.08 |
| | Steam ..... | 14.38 | 12.7 | 11.68 | 10.97 | 10.43 | 10.02 | 9.68 | 9.4 | 9.16 |
| 16 | Expansion... | 4.06 | 5.63 | 7.19 | 8.75 | 10.31 | 11.88 | 13.44 | 15 | 16.56 |
| | M.E.P..... | 38.43 | 43.64 | 47.54 | 50.71 | 53.33 | 55.59 | 57.57 | 59.33 | 60.91 |
| | Steam ..... | 14.71 | 12.95 | 11.79 | 11.15 | 10.6 | 10.17 | 9.82 | 9.53 | 9.28 |
| 17 | Expansion... | 3.82 | 5.29 | 6.76 | 8 24 | 9.71 | 11.18 | 12.64 | 14.12 | 15.59 |
| | M.E.P..... | 39.8 | 45.33 | 49.48 | 52.87 | 55.64 | 58.04 | 60.14 | 62.01 | 63.69 |
| | Steam ..... | 15.04 | 13.2 | 12.09 | 11 32 | 10.76 | 10.31 | 9.95 | 9.65 | 9.4 |
| 18 | Expansion... | 3.61 | 5 | 6.39 | 7.78 | 9.17 | 10.56 | 11.94 | 13.33 | 14.72 |
| | M.E.P..... | 41.11 | 46.97 | 51.36 | 54.92 | 57.88 | 60.42 | 62.64 | 64 66 | 66.08 |
| | Steam ..... | 15.36 | 13.44 | 12.29 | 11.5 | 10 91 | 10.45 | 10.08 | 9.76 | 9.51 |
| 19 | Expansion... | 3.42 | 4.73 | 6.05 | 7 37 | 8.68 | 10 | 11.32 | 12 63 | 13.95 |
| | M.E.P..... | 42.37 | 48.55 | 53.19 | 56.94 | 60.07 | 62 75 | 65.1 | 67.19 | 69.07 |
| | Steam ..... | 15.68 | 13.68 | 12.48 | 11.67 | 11.06 | 10.59 | 10.21 | 9.89 | 9 62 |
| 20 | Expansion... | 3.25 | 4 5 | 5.75 | 7 | 8.25 | 9.5 | 10.75 | 12 | 13.25 |
| | M.E.P..... | 43.57 | 50 08 | 54.96 | 58.92 | 62.2 | 65.03 | 67.5 | 69.7 | 71.68 |
| | Steam ..... | 16 | 13.92 | 12.68 | 11.84 | 11.21 | 10.72 | 10.33 | 10 | 9.72 |
| 21 | Expansion... | 3.09 | 4.29 | 5.48 | 6.67 | 7.86 | 9.05 | 10.24 | 11.42 | 12.62 |
| | M.E.P..... | 44.73 | 51.56 | 56.69 | 60.84 | 64.29 | 67.25 | 69.85 | 72.16 | 74.24 |
| | Steam ..... | 16.32 | 14 16 | 12.89 | 12 | 11.35 | 10.89 | 10.45 | 10.12 | 9.83 |
| 22 | Expansion... | 2.95 | 4.1 | 5.23 | 6.36 | 7.5 | 8.64 | 9.77 | 10.91 | 12.05 |
| | M.E.P..... | 45.83 | 52 99 | 58.36 | 62.71 | 66.33 | 69.43 | 72.15 | 74.57 | 76.75 |
| | Steam ..... | 16.64 | 14.39 | 13.06 | 12.16 | 11.5 | 10 98 | 10.57 | 10.22 | 9.93 |
| 23 | Expansion... | 2.83 | 3.19 | 5 | 6.09 | 7.17 | 8.26 | 9.35 | 10.43 | 11.52 |
| | M.E.P..... | 46.91 | 54.39 | 60 | 64 55 | 68.33 | 71.57 | 74.41 | 76.94 | 79.22 |
| | Steam ..... | 16.95 | 14.52 | 13.25 | 12.32 | 11.64 | 11.11 | 10 69 | 10.33 | 10.04 |
| 24 | Expansion... | 2.71 | 3.75 | 4.79 | 5.83 | 6.46 | 7 92 | 8.54 | 10 | 11.04 |
| | M.E.P..... | 47.91 | 55.72 | 61.57 | 66.33 | 70.27 | 73.66 | 76.62 | 79.26 | 81.64 |
| | Steam ..... | 17.27 | 14.85 | 13.44 | 12 47 | 11.78 | 11.28 | 10 8 | 10.44 | 10.14 |
| 25 | Expansion... | 2 6 | 3.6 | 4.6 | 5.6 | 6.6 | 7.6 | 8.6 | 9.6 | 10.6 |
| | M.E.P..... | 48.89 | 57.03 | 63.12 | 68.07 | 72.18 | 75.7 | 78 79 | 81.55 | 84.02 |
| | Steam ..... | 17.59 | 15.08 | 13.62 | 12 63 | 11.91 | 11.35 | 10 91 | 10.54 | 10.23 |
| 26 | Expansion... | 2.5 | 3.46 | 4.42 | 5.38 | 6 35 | 7.31 | 8.27 | 9.23 | 10.19 |
| | M.E.P..... | 49.82 | 58.28 | 64.62 | 69.77 | 74.11 | 77.71 | 80.92 | 83.79 | 86.36 |
| | Steam ..... | 17.9 | 15.27 | 13.8 | 12.79 | 12.03 | 11.48 | 11.02 | 10.65 | 10.33 |
| 27 | Expansion... | 2.47 | 3 33 | 4.26 | 5.18 | 6.11 | 7.04 | 7.96 | 8.89 | 9.81 |
| | M.E.P..... | 50.65 | 59 45 | 66.05 | 71.41 | 75 85 | 79.67 | 83.01 | 85.99 | 88.66 |
| | Steam ..... | 18 25 | 15.55 | 13.99 | 12.94 | 12.15 | 11.6 | 11.13 | 10.75 | 10.42 |
| 28 | Expansion... | 2.32 | 3.21 | 4.11 | 5 | 5.89 | 6.79 | 7.68 | 8.57 | 9.46 |
| | M.E.P..... | 51.58 | 60.69 | 67.52 | 73.01 | 77 66 | 81.62 | 85.08 | 88.16 | 90 93 |
| | Steam ..... | 18.54 | 15.76 | 14.16 | 13.1 | 12.31 | 11.72 | 11.24 | 10.85 | 10.52 |
| 29 | Expansion... | 2.24 | 3.1 | 3.97 | 4.83 | 5.69 | 6.55 | 7.41 | 8.28 | 9.14 |
| | M.E.P..... | 52.4 | 61.85 | 68.92 | 74.66 | 79.42 | 83.51 | 87.1 | 90.29 | 93.16 |
| | Steam ..... | 18.86 | 15.98 | 14.34 | 13.24 | 12.45 | 11.84 | 11.35 | 10.95 | 10.61 |
| 30 | Expansion... | 2.17 | 3 | 3.83 | 4.67 | 5.5 | 6.33 | 7.17 | 8 | 8.83 |
| | M.E.P..... | 53.2 | 62.96 | 70.28 | 76.22 | 81.15 | 85.38 | 89.1 | 92.39 | 95.36 |
| | Steam ..... | 19.18 | 16.2 | 14.51 | 13.39 | 12.57 | 11.95 | 11.45 | 11.04 | 10.7 |

## TABLE 8.—MEAN EFFECTIVE PRESSURE, AND STEAM CONSUMPTION PER I.H.P. PER HOUR.

*At a Steam Pressure 200 Pounds Above Atmosphere, 215 Pounds Absolute.*

| Terminal pressure. | At a card effect of— | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 100 per cent. | | | 70 per cent. | | 67½ per cent. | | 65 per cent. | | 60 per cent. | | 55 per cent. | |
| | Expansion. | M.E.P. | Steam pr. I.H.P. per hour. | M.E.P. | Steam pr. I.H.P. per hour. | M.E.P. | Steam pr. I.H.P. per hour. | M.E.P. | Steam pr. I.H.P. per hour. | M.E.P. | Steam pr. I.H.P. per hour. | M.E.P. | Steam pr. I.H.P. per hour. |
| 5 | 43.00 | 23.81 | 7.93 | 16.67 | 11 33 | 16.07 | 11.75 | 15.48 | 12.20 | 14.29 | 13 22 | 13.10 | 14.42 |
| 6 | 35.83 | 27.47 | 8.17 | 19.23 | 11.67 | 18.54 | 12.10 | 17.86 | 12.57 | 16.48 | 13.62 | 15.11 | 14.86 |
| 7 | 30.71 | 30.97 | 8.38 | 21.68 | 11.97 | 20.90 | 12.42 | 20.13 | 12.89 | 18.58 | 13.97 | 17.03 | 15.24 |
| 8 | 26.88 | 34.33 | 8 57 | 24.03 | 12.24 | 23.17 | 12.70 | 22.32 | 13.18 | 20.60 | 14.28 | 18.88 | 15.58 |
| 9 | 23.89 | 37.56 | 8.75 | 26.29 | 12.50 | 25.35 | 12.96 | 24.41 | 13.46 | 22.54 | 14.58 | 20.66 | 15.91 |
| 10 | 21.50 | 40.68 | 8.93 | 28 47 | 12.76 | 27.46 | 13.23 | 26 44 | 13.70 | 24.41 | 14.89 | 22.37 | 16.24 |
| 11 | 19.55 | 43.70 | 9.09 | 30.59 | 12.99 | 29 50 | 13.47 | 28.41 | 13.98 | 26.22 | 15.15 | 24.04 | 16.53 |
| 12 | 17.92 | 46.63 | 9.25 | 32.64 | 13.21 | 31.48 | 13.70 | 30.31 | 14.23 | 27.98 | 15.42 | 25.65 | 16 82 |
| 13 | 16.54 | 49.48 | 9.40 | 34.63 | 13.43 | 33.40 | 13.93 | 32.16 | 14.46 | 29 69 | 15.67 | 27.22 | 17.09 |
| 14 | 15.36 | 52.24 | 9.54 | 36.57 | 13.63 | 35.26 | 14.13 | 33.96 | 14.68 | 31.35 | 15.90 | 28.73 | 17.35 |
| 15 | 14.33 | 54.94 | 9.68 | 38.45 | 13.83 | 37.08 | 14.34 | 35.71 | 14.89 | 32.97 | 16 14 | 30.22 | 17.60 |
| 16 | 13.44 | 57.57 | 9.82 | 40.30 | 14.03 | 38.86 | 14.55 | 37.43 | 15.11 | 34.54 | 16.37 | 31.66 | 17 86 |
| 17 | 12.64 | 60.14 | 9.95 | 42.10 | 14.22 | 40.60 | 14.74 | 39.09 | 15.31 | 36.09 | 16.58 | 33.08 | 18.09 |
| 18 | 11.94 | 62.64 | 10.08 | 43.85 | 14.40 | 42.28 | 14.93 | 40.72 | 15.51 | 37.59 | 16.80 | 34 45 | 18.33 |
| 19 | 11.32 | 65.10 | 10.21 | 45.57 | 14.58 | 43.94 | 15.13 | 42 32 | 15.71 | 39.06 | 17.02 | 35.81 | 18.56 |
| 20 | 10.75 | 67.50 | 10.33 | 47.25 | 14.76 | 45.57 | 15.30 | 43.88 | 15.89 | 40.50 | 17.22 | 37.12 | 18.78 |
| 21 | 10.24 | 69.85 | 10.45 | 48.88 | 14.93 | 47.14 | 15.48 | 45.41 | 16.08 | 41.91 | 17.42 | 38.42 | 19.00 |
| 22 | 9.77 | 72.15 | 10.57 | 50.50 | 15,10 | 48.70 | 15.66 | 46.90 | 16.26 | 43.29 | 17.62 | 39.68 | 19.22 |
| 23 | 9.35 | 74.41 | 10.69 | 52.08 | 15.27 | 50.22 | 15.84 | 48 37 | 16 45 | 44 65 | 17.82 | 40.93 | 19.43 |
| 24 | 8.54 | 76.62 | 10.80 | 53.63 | 15.43 | 51.72 | 16.00 | 49.81 | 16.61 | 45.97 | 18 00 | 42.14 | 19 64 |
| 25 | 8.60 | 78.79 | 10.91 | 55.15 | 15.59 | 53.18 | 16.16 | 51.22 | 16,78 | 47.28 | 18.18 | 43.34 | 19.84 |
| 26 | 8.27 | 80.92 | 11.02 | 56.64 | 15.74 | 54.62 | 16.33 | 52.60 | 16.95 | 48,56 | 18.37 | 44.51 | 20.22 |
| 27 | 7.96 | 83.01 | 11.13 | 58.10 | 15.90 | 56.03 | 16.49 | 53 96 | 17.12 | 49 81 | 18.55 | 45.66 | 20.24 |
| 28 | 7.68 | 85.08 | 11.24 | 59.55 | 16.06 | 57.43 | 16.65 | 55.31 | 17.29 | 51.05 | 18.73 | 46.80 | 20.44 |
| 29 | 7 41 | 87.10 | 11.35 | 60.97 | 16.21 | 58.79 | 16.82 | 56.62 | 17.46 | 52.26 | 18.92 | 47.86 | 20.64 |
| 30 | 7.17 | 89.10 | 11.45 | 62.37 | 16.36 | 60.14 | 16.96 | 57.92 | 17.61 | 53.46 | 19.08 | 49.01 | 20 82 |

TABLE 9.—MEAN EFFECTIVE PRESSURE, AND STEAM CONSUMPTION PER I.H.P. PER HOUR.

*At a Steam Pressure 225 Pounds Above Atmosphere, 240 Pounds Absolute.*

| Terminal pressure. | At a card effect of— | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 100 per cent. | | | 70 per cent. | | 67½ per cent. | | 65 per cent. | | 60 per cent. | | 55 per cent. | |
| | Expansion. | M.E.P. | Steam pr. I.H.P. per hour. | M.E.P. | Steam pr. I.H.P. per hour. | M.E.P. | Steam pr. I.H.P. per hour. | M.E.P. | Steam pr. I.H.P. per hour. | M.E.P. | Steam pr. I.H.P. per hour. | M.E.P. | Steam pr. I.P.H. per hour. |
| 5 | 48.00 | 24.36 | 7.75 | 17 05 | 11.07 | 16.44 | 11.48 | 15.83 | 11.92 | 14.62 | 12.92 | 13.40 | 14.09 |
| 6 | 40.00 | 28.13 | 7 98 | 19.69 | 11.40 | 18.99 | 11.82 | 18.28 | 12.28 | 16.88 | 13.30 | 15.47 | 14.51 |
| 7 | 34.29 | 31.74 | 8.18 | 22.22 | 11.69 | 21.42 | 12.12 | 20.63 | 12.58 | 19.05 | 13.63 | 17.46 | 14.87 |
| 8 | 30.00 | 35.21 | 8.36 | 24 64 | 11.94 | 23.77 | 12.39 | 22.89 | 12.86 | 21.13 | 13.93 | 19.37 | 15.20 |
| 9 | 26.67 | 38.50 | 8.54 | 26.95 | 12.20 | 25.99 | 12.65 | 25.03 | 13.14 | 23.10 | 14.23 | 21.17 | 15.53 |
| 10 | 24 00 | 41.78 | 8.69 | 29.25 | 12.42 | 28.20 | 12 88 | 27.16 | 13.37 | 25.07 | 14 48 | 22.98 | 15 80 |
| 11 | 21 82 | 44 92 | 8.84 | 31.44 | 12.63 | 30.32 | 13.10 | 29.20 | 13.60 | 26.95 | 14.73 | 24.71 | 16.07 |
| 12 | 20.00 | 47.95 | 8.99 | 33.56 | 12.84 | 32.37 | 13.32 | 31.17 | 13.83 | 28.77 | 14.98 | 26.37 | 16.34 |
| 13 | 18.46 | 50 91 | 9.13 | 35.63 | 13.04 | 34.37 | 13.53 | 33.09 | 14.05 | 30.55 | 15.22 | 28.00 | 16.60 |
| 14 | 17.14 | 53.74 | 9.28 | 37.62 | 13.26 | 36.27 | 13.75 | 34.93 | 14.28 | 32.25 | 15.47 | 29.56 | 16.87 |
| 15 | 16.00 | 56.59 | 9.40 | 39.61 | 13.43 | 38.20 | 13.93 | 36.78 | 14.46 | 33.95 | 15.67 | 31.12 | 17.09 |
| 16 | 15.00 | 59.33 | 9.53 | 41.53 | 13.62 | 40.05 | 14.12 | 38.57 | 14.66 | 35.60 | 15.88 | 32.63 | 17.33 |
| 17 | 14.12 | 62.01 | 9.65 | 43.40 | 13.79 | 41.86 | 14.30 | 40.31 | 14.85 | 37.21 | 16.08 | 34.11 | 17.54 |
| 18 | 13.33 | 64.66 | 9.76 | 45.26 | 13 94 | 43.65 | 14.46 | 42.03 | 15.01 | 38 80 | 16.27 | 35.56 | 17.74 |
| 19 | 12.63 | 67.19 | 9.89 | 47.03 | 14.13 | 45.35 | 14.65 | 43.68 | 15.22 | 40.31 | 16.48 | 36.95 | 17.98 |
| 20 | 12.00 | 69.70 | 10.00 | 48.79 | 14.29 | 47.05 | 14.82 | 45.31 | 15.38 | 41.82 | 16.67 | 38.34 | 18.18 |
| 21 | 11.42 | 72.16 | 10.12 | 50.51 | 14.46 | 48.71 | 14.99 | 46.91 | 15.57 | 43.30 | 16.87 | 39.68 | 18.40 |
| 22 | 10.91 | 74.57 | 10.22 | 52.19 | 14.60 | 50.33 | 15.14 | 48.47 | 15.72 | 44 74 | 17.03 | 41.01 | 18.58 |
| 23 | 10.43 | 76.94 | 10.33 | 53.86 | 14.76 | 51.93 | 15.30 | 50.01 | 15.89 | 46.17 | 17.22 | 42.32 | 18.78 |
| 24 | 10.00 | 79.26 | 10.44 | 55.48 | 14.91 | 53.50 | 15.47 | 51.52 | 16 06 | 47.56 | 17.40 | 43.59 | 18.98 |
| 25 | 9.60 | 81.55 | 10.54 | 57.08 | 15.06 | 55.04 | 15.62 | 53.01 | 16.21 | 48.93 | 17.57 | 44.85 | 19.16 |
| 26 | 9.23 | 83 79 | 10.65 | 58.65 | 15.21 | 56.56 | 15.78 | 54.46 | 16.38 | 50.27 | 17.75 | 46.09 | 19 36 |
| 27 | 8.89 | 85.99 | 10.75 | 60.19 | 15 36 | 58.04 | 15.93 | 55.89 | 16.54 | 51.59 | 17.92 | 47.30 | 19 54 |
| 28 | 8.57 | 88.16 | 10.85 | 61.72 | 15 50 | 59.51 | 16.07 | 57.31 | 16.69 | 52.90 | 18.08 | 48.49 | 19.72 |
| 29 | 8.28 | 90.29 | 10.95 | 63.20 | 15.64 | 60.94 | 16.22 | 58.69 | 16.85 | 54.17 | 18.25 | 49.66 | 19.91 |
| 30 | 8.00 | 92.39 | 11.04 | 64.67 | 15.77 | 62.36 | 16.36 | 60.05 | 16.98 | 55.44 | 18.40 | 50.81 | 20.09 |

## TABLE 10.—MEAN EFFECTIVE PRESSURE, AND STEAM CONSUMPTION PER I.H.P. PER HOUR.

*At a Steam Pressure 250 Pounds Above Atmosphere, 265 Pounds Absolute.*

| Terminal pressure. | At a card effect of— | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 100 per cent. | | | 70 per cent. | | 67½ per cent. | | 65 per cent. | | 60 per cent. | | 55 per cent. | |
| | Expansion. | M.E.P. | Steam pr. I.H.P. per hour. | M.E.P. | Steam pr. I.H.P. per hour. | M.E.P. | Steam pr. I.H.P. per hour. | M.E.P. | Steam pr. I.H.P. per hour. | M.E.P. | Steam pr. I.H.P. per hour. | M.E.P. | Steam per I.H.P. per hour. |
| 5 | 53.00 | 24.85 | 7.60 | 17.40 | 10.86 | 16.77 | 11.26 | 16.15 | 11.69 | 14.91 | 12.67 | 13.67 | 13.82 |
| 6 | 44.17 | 28.73 | 7.81 | 20.11 | 11.16 | 19.39 | 11.57 | 18.67 | 12.02 | 17.24 | 13.02 | 15.80 | 14.15 |
| 7 | 37.86 | 32.44 | 8.00 | 22.71 | 11.43 | 21.90 | 11.85 | 21.09 | 12.31 | 19.46 | 13.33 | 17.84 | 14.54 |
| 8 | 33.13 | 36.00 | 8.17 | 25.20 | 11.67 | 24.30 | 12.10 | 23.40 | 12.57 | 21.60 | 13.62 | 19.80 | 14 85 |
| 9 | 29.44 | 39.44 | 8.34 | 27.61 | 11.91 | 26.62 | 12.35 | 25.64 | 12.83 | 23.66 | 13.90 | 21.69 | 15.16 |
| 10 | 26.50 | 42.77 | 8.49 | 29.94 | 12.13 | 28.87 | 12.58 | 27.80 | 13.06 | 25.66 | 14.15 | 23.52 | 15.44 |
| 11 | 24.10 | 46.01 | 8.63 | 32.21 | 12.33 | 31.05 | 12.78 | 29 91 | 13.28 | 27.61 | 14.38 | 25.31 | 15.69 |
| 12 | 22.08 | 49.14 | 8.78 | 34.40 | 12.54 | 33.17 | 13.01 | 31.94 | 13.51 | 29 49 | 14.63 | 27.03 | 15.96 |
| 13 | 20.38 | 52.19 | 8 91 | 36.54 | 12.73 | 35.23 | 13.20 | 33.93 | 13.71 | 31.31 | 14.85 | 28.70 | 16.20 |
| 14 | 18.93 | 55.17 | 9.03 | 38.62 | 12.90 | 37.24 | 13.38 | 35. 86 | 13 89 | 33.10 | 15.05 | 30.34 | 16.42 |
| 15 | 17.67 | 58.08 | 9.16 | 40.66 | 13.09 | 39.20 | 13.57 | 37.75 | 14.09 | 34.85 | 15.27 | 31.94 | 16.65 |
| 16 | 16.56 | 60.91 | 9.28 | 42.64 | 13 26 | 41.11 | 13.75 | 39.59 | 14.28 | 36.55 | 15.47 | 33.50 | 16.87 |
| 17 | 15.59 | 63.69 | 9.40 | 44.58 | 13.43 | 42.99 | 13.92 | 41.40 | 14.46 | 38.22 | 15.67 | 35.03 | 17.09 |
| 18 | 14.72 | 66.08 | 9.51 | 46.26 | 13.59 | 44.60 | 14.09 | 42.95 | 14.63 | 39.65 | 15.85 | 36.35 | 17.29 |
| 19 | 13.95 | 69.07 | 9.62 | 48.35 | 13.74 | 46.62 | 14.25 | 44.90 | 14.80 | 41.44 | 16.03 | 37.99 | 17.49 |
| 20 | 13.25 | 71.68 | 9.72 | 50 18 | 13.89 | 48.38 | 14.40 | 46.59 | 14.95 | 43.01 | 16.20 | 39.42 | 17.67 |
| 21 | 12.62 | 74.24 | 9.83 | 51.97 | 14 04 | 50.11 | 14.56 | 48.26 | 15.12 | 44.55 | 16.38 | 40.83 | 17.87 |
| 22 | 12.05 | 76.75 | 9.93 | 53.73 | 14 19 | 51.80 | 14.71 | 49.88 | 15.28 | 46 05 | 16.55 | 42.21 | 18.06 |
| 23 | 11.52 | 79.22 | 10.04 | 55.46 | 14.34 | 53.47 | 14.87 | 51.50 | 15.45 | 47.53 | 16.73 | 43.57 | 18.25 |
| 24 | 11.04 | 81.64 | 10.14 | 57.15 | 14.48 | 55.10 | 15.02 | 53.07 | 15.60 | 48.99 | 16.90 | 44.90 | 18.44 |
| 25 | 10.60 | 84.02 | 10.23 | 58.82 | 14 61 | 56.71 | 15.15 | 54.61 | 15.74 | 50.41 | 17.05 | 46.21 | 18.60 |
| 26 | 10.19 | 86.36 | 10.33 | 60.45 | 14.76 | 58.28 | 15.30 | 56.14 | 15.89 | 51.82 | 17.22 | 47.50 | 18.78 |
| 27 | 9.81 | 88.66 | 10.42 | 62.07 | 14.89 | 59.84 | 15.43 | 57.63 | 16.03 | 53.20 | 17.37 | 48.76 | 18.94 |
| 28 | 9.46 | 90.93 | 10.52 | 63.65 | 15.03 | 61.38 | 15.58 | 59.11 | 16.19 | 54.56 | 17.53 | 50.01 | 19.13 |
| 29 | 9.14 | 93.16 | 10.61 | 65.22 | 15.16 | 62.88 | 15.72 | 60.56 | 16.32 | 55.90 | 17.68 | 51.24 | 19.29 |
| 30 | 8.83 | 95.36 | 10.70 | 66.76 | 15.29 | 64.36 | 15.85 | 61.98 | 16.48 | 57.22 | 17.83 | 52.45 | 19.45 |

# ADVANCED ENGINEERING EDUCATION.

ADDRESS OF PROF. CHAS. LÜEKE,

*At the Banquet A. S. N. E., April 25, 1914.*

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It is the custom in courts of law to collect evidence from witnesses on which to base a judgment. In complex cases witnesses of two sorts are recognized, ordinary witnesses who may testify only to facts known by them at first hand, and expert witnesses who by reason of training and experience may be qualified to express conclusions on facts, or to interpret them. Assuming that this gathering is interested in the case of the post-graduate school of engineering, I have come to offer evidence both as an ordinary and as an expert witness. My right to speak as an ordinary witness is based on a year of teaching and observation of two classes of men of lieutenant grade, one starting its first year at Annapolis, and the other completing its second year at Columbia University, both engaged in studying the engineering of the machinery of a ship. My authority for such expert interpretation and opinions as I shall offer is to be found in my own experience as a practicing engineer engaged almost continuously in professional engineering practice and research, and as professor of mechanical engineering in the largest University in America, training several thousand young engineers for all branches of professional practice.

The net conclusion of the year's work is that the post-graduate school as at present constituted is an unquestionable success, and as such entitled to serious consideration, not only from the Navy as a whole for the direct assistance to its work that must result, but also from the country at large which de-

mands that the American Navy be second to none in efficiency, equipment and personnel, just as American financial resources, industrial advance and engineering practice are second to none. In fact, it can be demonstrated that it is precisely this engineering efficiency that is so largely responsible for our present economic strength, and exactly the same thing that is needed to make naval equipment all it might be and should be.

It is quite worth while to examine the conditions a little more closely that they may be more clearly understood and improvements planned, for the best of things today may be made better tomorrow. The purpose of any sort of training for naval officers must be for effectiveness of the Navy as a unit, or its officers individually, in fact, both. The sort of training required must, of course, be based on the subject and on the object, which in this case resolve into, first, the sort of man to be trained and his previous experience, and second, the equipment that must be produced and handled.

The modern fighting ship is a great intricate and costly mechanical organism, performing harmoniously a most amazingly complex series of physical, chemical and mechanical operations and processes. Its vital organs consist of a cooperating series of individual machines and pieces of apparatus, each designed for a specific purpose, each one alone, and all together as a unit, produced by human skill, and subject to human direction for their utility. There is no single structure or aggregation to be found anywhere in the world that is at once so complex and varied, or that requires so much highly developed scientific and practical knowledge for both original design, for construction, and for subsequent economic operation and maintenance. Nor is there even in this country of great industrial organizations, established and continued by conceptions of the brain of the engineer, a single one as great as the Navy Department itself, which must not only produce the best to be had in fighting ships, the best shops to build and repair them, but also must supply skilled men to create, use and care for them.

## NECESSITY FOR SPECIALIZATION.

It is utterly beyond the capacity of any one man to acquire in a lifetime the complete mastery of all the pertinent physical processes, apparatus, machinery and materials to the degree that modern scientific engineering thoroughness demands. It is still less possible for any one man to become at the same time skilled in the art of war. It, therefore, follows that the aggregation of men constituting the Navy organization must itself be modeled somewhat on the lines of the ship's mechanism itself, where the function of support and enclosure is performed by hull and decks, those of propulsion and habitability by main and auxiliary machinery, that of defense by ordinance, and of communication by signalling equipment, but all so harmoniously coöperating that not only one ship, but a whole fleet may act as one unit. That an organization of men may efficiently act as one unit while performing within itself a vastly complex system of functions, there has been established among the engineering industries the principle of coöperative division of labor, according to which every man must learn something of the duties of all, but become highly skilled in at least one, the one best suited to his natural ability and which duty he may himself perform, or, on which, act as instructor, director, or as a court of last resort. As the knowledge and skill of the world increases more and more sub-divisions of function must be created because of the growing sum total, and more and more differently endowed or trained men efficiently used, each man skilled in a constantly lessening percentage of the total, and this must go on forever.

Such a condition of affairs is characteristic of engineering practice in which not longer than fifty years ago one man then termed a Civil Engineer could be skilled in all there was of engineering knowledge, while today there is generally recognized the broad subdivisions of mechanical engineer, skilled in creating and using machinery; mining engineer, in mineral location and recovery; metallurgical engineer, in metal extraction and refinement; chemical engineer, in production of

new substances from old; electrical engineer, in current generation and use, and civil engineer, in static structures and public works. Each of these now has its further subdivisions, such as heating and ventilating, refrigeration, gas, automobile, locomotive, marine engineering and industrial production or efficiency engineering, the number constantly and necessarily increasing; and the end will never come.

## THE FIELD OF NAVAL ENGINEERING.

My own professional interest in the Navy organization and equipment is concentrated on its main and auxiliary machinery and the human means whereby the best that mechanical engineering has may be made available and be economically used, properly maintained and repaired. On this I feel entitled to speak with authority. The mechanical equipment for converting the heat of combustion of coal and oil fuel to power to drive the ship, to pump water within it, to refrigerate food, to heat and ventilate quarters, to distill fresh from salt water, to generate electric current for illumination and electric motors, is itself surprisingly complex and so rapidly improving and changing in kind that the best of one generation is useless scrap in the next. The present breadth and depth of engineering knowledge is such that it is not possible for one man to keep fully informed on the development of new principles of operation and construction of this whole group of machinery. There is so much known and to be discovered on any one part or sub-class as to require all the mental capacity and time of any man who would be an authority on it. Accordingly, not only do the facts justify the complete attention of one group of men on the subject of main and auxiliary machinery, but they, to an even more rational degree, warrant individual devotion to at least one of each of such separate topics as boilers, fuels, coal furnaces, oil burners, superheaters, steam turbines, reciprocating steam engines, pumps for boiler feed, condenser, bilge and circulation pumps, fans, valves, piping and ducts, heating systems, refrigerating systems, feed-

water heaters and condensers, as well as other advancing subjects like the gasoline engine for small boats, Diesel and equivalent type oil engine for larger vessels, and gas engines and gas producers, which are being studied and perfected for shore use and which may be adapted to Navy requirements.

All of these things must be created by design and construction, and then be economically operated and effectively maintained and repaired, as stated before. The latter two functions are fundamentally Navy Department duties, while the former—design and construction—may be or may not be. If not, then the substitute is purchase from civilian manufacturers instead of Bureau design and navy-yard construction. However, even in the latter case of purchase, specification must originate in the Department, and without adequate engineering knowledge of what may be had, which knowledge presupposes ability to produce it independently, proper specification is impossible, and is likely to range from a demand for something that is not the best to something so impossibly good as to be unattainable short of heaven itself.

It is in such facts as these here so briefly presented that proof is to be found of the proposition that engineering knowledge is an absolute necessity to modern naval efficiency.

## SOURCES OF ENGINEERING KNOWLEDGE.

Such engineering knowledge as is necessary cannot be gained without contact with the sources of that knowledge, which assuredly do not reside within a ship. The men whose sole duty is to develop, create, analyze their own product and that of others, and unceasingly to improve machinery, are themselves the sources of such information, and it is from them that it must be learned. To expect such to be gained by operating the machinery of a vessel is just about as reasonable or unreasonable as for us to expect to train architects by laying bricks, or to train designers, manufacturers or purchasers of locomotives by riding in a cab, of boilers by shoveling coal, or of turbines by turning on or off steam; in short,

it is utterly hopeless. Men to do these things must be trained in those engineering principles that are the summation of the history of engineering development. This body of engineering principles is the only safe and sane guide to the solution of present-day and future problems, and their understanding is as necessary to the engineer charged with these problems as a knowledge of the principles of the law to the court in judging questions of human rights and to the legislator who must adapt, modify and extend them to meet the demands that come with the march of time. Engineering principles are classified according to typical natural laws, stable, unvarying and clarifying; gravitation, which is one of them, said to have been discovered by Newton, observing the fall of an apple from a tree, will serve to illustrate the point. It is now the basis of all explanation and prediction of a wonderful series of phenomena ranging from the movement of the heavenly bodies to the velocity of steam through the nozzle of a turbine engine. They become the invisible commanders of the designer, who must know the force deforming every element of his structure and the resisting value of every material of construction; the heating surface and material of a boiler to transmit without injury enough heat from a fire to make the required steam, the size and shape of the cylinders of his engine to develop the required power with the use of a limited amount of steam. They are the friends of the operator who would get the maximum effect from a pound of coal burned, or in emergency attain an extraordinary speed, or keep the ship supplied with fresh water with an injured evaporator, or avoid breakdowns by pointing out an overstressed rod or shaft so that it may be more carefully guarded by extra precautionary care.

## EFFICIENT SPECIAL TRAINING ESSENTIAL.

Principles of this sort are not learned without much time and effort, even under the most favorable circumstances, and it is safe to say that they cannot be truly and efficiently learned at all except by a course of training from men who have made

them a special study and who know how to impart what they know, and then only by students of some maturity and, preferably, of some practical experience in the chosen field of application. Training must not cease with the demonstration of the principles of engineering; it must include drill in its methods and the mode of application to specific things, and, finally, last but not least, a development of love and respect for that most subtle but none the less real and useful thing, its spirit or ideal. The method of engineering is the method of all science analysis, synthesis, comparison, generalization, codification and orderly mental procedure; its spirit and ideal is likewise common to all science—a rigid adherence to abstract and impersonal truth and perpetual dissatisfaction with anything short of perfection. There *may be* other and better ways of training your young men in these principles, methods and ideals and their application to the problems of naval machinery than the present post-graduate plan, which training must inevitably make not only better engineers but better men, as men and as officers, as surely as any ethics or national patriotism, but if there be any such better ways I have not heard of them. I do know that the present plan is successfully workable and can be made satisfactory for future needs. Both the need of the training and the possibility of imparting it by the present system are amply demonstrated to my satisfaction by the experience of this year, and I heartily wish there were sufficient time available to lay before you in detail the array of facts that would support this conclusion.

## RESULTS OF CO-ORDINATION AND INVESTIGATION.

A few broad facts stand out so clearly, however, as to warrant their statement. The relative contributions of the strictly naval members of the school staff with those of the University members have been effectively coördinated in a manner satisfactory to both, which is good news, as there might have easily been conflict. The student officers have shown an attention to duty, a cheerfulness and fixity of pur-

pose in the fact of great difficulties that is deserving of the highest praise.

In all my experience I have never directed the work of a finer body of men as men, and the Navy may well be proud of the success of its training along these lines. Their capacity for memorizing is excellent—in my opinion, excessively developed, almost to a photographic degree. Their analytical power and their capacity for reasoning and their knowledge of engineering principles, breadth of view and grasp of the professional engineering ideals of impersonal values are, however, all in serious need of development, and this their present training is now accomplishing; minor faults have been reduced and in some cases clearly eliminated, so it is possible to report a marked improvement. One man who had been in charge of boilers for several years said that a boiler would never look the same again; it had now become a thing endowed with life and life interest instead of an inanimate structure. Another, reporting no knowledge of the standard shore system of refrigeration and no previous interest because it was not approved for the Navy, is now occupied with ways and means for adaptation. All were surprised and somewhat indignant, at first, to discover that of two boilers whose tests were available, the one that showed the highest overall efficiency and so seemed to be better, really was not, but really less efficient, analysis proving it to have had a lower heat-absorbing efficiency and higher furnace efficiency than the other; change of furnaces would have resulted in a combination better than either possessed originally. These are typical stories of facts and experiences, of which I could give you hundreds; they all point the same way. The present system of post-graduate instruction is good, and by proper development is bound to become a great force for good in the Navy. This development I venture to suggest should involve further perfection of the details of the course of study, more active coöperation of the Bureau of Steam Engineering, development of contact with the problems of the fleet as well as those of the Bureau, and definite provision for research to follow instruction.

Finally, let me recall a remark made by one you all know, Sir William White, who, after commending American mechanical engineering papers for their clear, fearless and purely professional discussion of present-day problems, regretted that in England the author opened his paper with a review of conditions at the time of the Egyptian pyramids, then followed it by carefully analyzing progress almost up to date—always almost, but never quite. Such English writers would gather the threads of all existing knowledge on the subject treated, then would never use it. They had a definite objective, they prepared for it, but never reached it. There is in this a lesson for us, let us by all means train our men in engineering principles and in naval engineering, but, for the sake of all we hold most dear, let us not stop with the training, but give the men an opportunity for such drill and experience as will firmly fix it, and provide chances to use it, so that the Service may benefit more and more without end, until our ships are always provided with something a little better than any others, and that, better handled; in short, until we have in quality the best in the world, with all others following our lead.

# DESCRIPTION AND TRIALS OF TORPEDO-BOAT DESTROYER *DUNCAN*.

By A. Conti, Associate.

The *Duncan* belongs to the class of destroyers Nos. 43 to 50, which forms part of the 1911–12 building program authorized by Congressional Act approved March 4, 1911.

The vessel was awarded to the Fore River Shipbuilding Corporation on December 6, 1911, at the contract price of $779,450.00 (exclusive of armament, but including installation of same), and the stipulated time of delivery was 24 months from date of contract. The contract calls for a twin-screw vessel of about 1,010 tons displacement and 29 knots speed, to be built upon hull-type plans furnished by the Navy Department and with contractors' design of propelling machinery.

## HULL.

The arrangement of quarters, armament and equipment is substantially the same for all the boats of this class. For a description of these items, reference may be had to an article on the U. S. S. *Aylwin, Parker* and *Benham,* which appeared in the February, 1914, number of the Journal.

It is sufficient to recall the main hull characteristics, which are as follows:

| | |
|---|---|
| Length on designed W.L., feet and inches | 300–00 |
| over all, feet and inches | 305–03 |
| Beam on designed W.L., molded, feet and inches | 30–02½ |
| extreme, molded, feet and inches | 30–05¾ |
| Designed draught, forward, feet and inches | 8–06⅝ |
| aft, feet and inches | 9–11⅝ |
| mean, feet and inches | 9–03⅛ |

| | |
|---|---|
| Displacement (including 205 tons of fuel oil and 12.3 tons of reserve feed water), tons | 1013.5 |
| Block coefficient | .419 |
| Midship section coefficient | .677 |
| Coefficient at fineness of D.W.L. at designed draught | .662 |
| Displacement per inch at D.W.L., tons | 14.28 |
| Wetted surface, square feet | 9,310 |
| Capacity of fuel oil tanks 95 per cent. full, tons | 305 |
| reserve feed-water compartments, tons | 34.7 |

## PROPELLING MACHINERY.

The propelling machinery consists of two sets of main turbines and compound, reciprocating, cruising engines, which are used in series with the main turbines for cruising speeds up to 15½ knots. The turbines are of the Curtis marine type, of 63 inches pitch diameter and 42 ahead stages, with a total of 51 rows of moving blades, and 10 stages with 14 rows of

TABLE II.—CRUISING-ENGINE DATA.

| | | | | |
|---|---|---|---|---|
| Two vertical, inverted, direct-acting, compound, reciprocating engines, 12½-inch × 26½-inch × 13½-inch stroke. | | | | |
| Length of connecting rods, 33 inches. Diameter of piston rods, 2¾ inches. R.P.M., 254. Piston speed, 571.5 feet per minute. | | | | |
| Cylinder | H.P. | | L.P. | |
| Number and size of piston valves, diameter, ins. | One, 7½ | | Two, 7½ | |
| Steam taken | Inside. | | Outside. | |
| Valve travel, inches | 3⅜ | | 3⅜ | |
| Width of steam ports in valve liners, inches | 1 5/16 | | 1 1/16 | |
| | Top. | Bottom. | Top. | Bottom. |
| Piston clearance, inch | 3/32 | 1/8 | 3/32 | 1/8 |
| Steam lap, inch | 1 5/16 | 3 1/2 | 1/8 | 1 1/16 |
| Exhaust lap, inch | 0 | +1/8 | 0 | +3/32 |
| Angular advance, degrees and minutes | 40–35 | | 48–20 | |
| Steam lead, linear, inch | 5/32 | 1/4 | 1/8 | 7/64 |
| Cut-off in decimal of stroke | .683 | .617 | .645 | .567 |
| mean, in decimal of stroke | .65 | | .606 | |
| Compression, in decimal of stroke | .142 | .119 | .193 | .155 |
| Exhaust release, in decimal of stroke | .902 | .885 | .862 | .830 |
| Steam opening, linear, inch | 1/4 | 27/32 | 1 1/16 | 7/8 |
| area, square inches | 11.72 | 13.19 | 28.94 | 31.18 |
| Exhaust opening, linear | Full | port | Full | port. |
| area, square inches | 14.66 | 14.66 | 37.86 | 37.86 |
| Steam velocity through valve-chest liner | 5,983 | 5,060 | 10,690 | 9,810 |
| Exhaust velocity through valve-chest liner | 4,783 | 4,553 | 8,170 | 8,080 |
| Velocity through main steam pipe, 3½-inch diameter | 7,290 | 6,953 | ... | ... |
| Velocity through main exhaust pipe, 10-inch diameter | ... | ... | 3,938 | 3,897 |

blades for going astern. Each turbine is designed to develop 7,500 S.H.P. at 550 r.p.m., corresponding to a speed of vessel of 29 knots; the designed pressure range being from 250 pounds gage at the turbine steam chest to an absolute pressure of 1.5 pounds at the exhaust shell. Forward of each turbine, and connected to it by means of a jaw clutch, is a 12½-inch by 26¼-inch by 13½-inch stroke vertical, inverted, direct-acting, compound reciprocating engine. When connected for use, this engine takes live steam at the H.P. valve chest, and exhausts practically at atmospheric pressure, into the first-stage chamber of the main turbine. The cruising engine can also be used for backing, as the valve gear is fitted for astern motion, the exhaust steam being led to the first-stage shell of the astern turbine by means of a change valve that takes its motion from the reversing shaft.

### SHAFTING AND BEARINGS.

There are two lines of shafting, each line being about 122 feet long over all.

All shafting is of Class "A" steel forgings, a few details being given in the following table:

| Name of section. | Shafting. | | | Bearings. | | | |
|---|---|---|---|---|---|---|---|
| | Length. | Diameter. | Bore. | Number. | Diameter. | Length. | Type. |
| | *Ft. and inches.* | *Ins.* | *Ins.* | | *Ins.* | *Ft. and inches.* | |
| Cruising engine, crank shaft | 8–0⅝ | 7½ | 3¼ | 4 | 7½ | 0–07 | Babbitted. |
| Turbine shaft | 21–6 | 10¼* | 8½ | 2 | 13 | 0–18 | Babbitted. |
| Line shaft, forward section | 11–4⅛ | 10 | 7 | None. | ... | ... | ..... |
| Line shaft, after section | 23–6½ | 10 | 7 | 2 | 10½ | 0–13½ | Babbitted. |
| Stern tube shaft | 24–7½ | 10 | 7 | 1 | 10½ | 2–2 | Lignum vitæ. |
| | | | | 1 | 10½ | 3–7½ | Lignum vitæ. |
| Propeller shaft | 32–2½ | 10 | 7 | 1 | 10½ | 2–7 | Lignum vitæ. |
| | | | | 1 | 10½ | 3–4½ | Lignum vitæ. |

* Maximum.

The main thrust bearing is at the forward end of the turbine shaft, and consists of four horseshoes of the ordinary type, babbitted on both faces and cored for the circulation of cooling water.

As the turbine is designed so that the steam thrust always overbalances the propeller thrust, the resultant thrust is directed aft, and the forward side of the horseshoes is the working face. The total effective surface of the four horseshoes, after deduction for oil grooves, is 260 square inches.

| | |
|---|---|
| Number of collars | 5 |
| Thickness, inches | $1\frac{1}{16}$ |
| Space between collars, inches | 3 |
| Outside diameter, inches | $14\frac{5}{8}$ |
| Inside diameter, inches | $8\frac{3}{4}$ |

The shaft horsepower developed by the main propelling machinery is measured by an improved type of Gary-Cummings torsionmeter, one for each line of shafting.

The torsionmeter constants were found by calibration of the actual shafts with the torsionmeters mounted on them, as described in the May, 1913, number of the Journal. The twist of the shaft was measured by means of pointers having a radius of eight feet and spaced six feet apart, enabling thus to figure the torsional modulus of elasticity.

The following were the results of the calibration:

| Line shaft | Starboard. | Port. |
|---|---|---|
| Length of shaft calibrated, feet and inches | 11–$4\frac{1}{16}$ | 11–$4\frac{1}{16}$ |
| Length of shaft which torsionmeters record, ft. and ins. | 10–$9\frac{13}{64}$ | 10–$9\frac{9}{16}$ |
| M = Twisting moment in foot pounds | 90,000 | 91,500 |
| T = Distance in inches between torsionmeter pointers, corresponding to M, corrected for zero reading | 2.90 | 2.94 |
| $\frac{M}{T}$ = Twisting moment in foot pounds per inch of meter reading | 31,034.48 | 31,122.45 |
| Meter constant K* $= \frac{2\pi \times \frac{M}{T}}{33{,}000}$ | 5.9089 | 5.9257 |
| Torsional modulus of elasticity | 11,834,000 | 11,801,300 |
| Twist of shaft for one inch movement of meter, degrees | .17411 | .17553 |

## PROPELLERS.

There are two three-bladed outboard-turning propellers. They are of manganese bronze, cast solid. The blades are true screw, machined to pitch and polished. The edges are dressed

*S.H.P. = constant K × distance in inches between marks on meter card × r.p.m

to the designed contour and filed sharp. The following are the actual propeller measurements:

| | Starboard. | Port. |
|---|---|---|
| Diameter, feet and inches | 7–11⅞ | 7–11¾ |
| Pitch, feet and inches | 6–06½ | 6–06 3/16 |
| Ratio pitch to diameter | 1.221 | 1.224 |
| Projected area, square feet | 24.96 | 25.14 |
| Helicoidal area, square feet | 28.66 | 28.98 |
| Disc area, square feet | 50.134 | 50.004 |
| Ratio projected to disc | .498 | .502 |
| helicoidal to disc | .572 | .579 |
| projected to helicoidal | .872 | .867 |
| Lower tip of blade above keel, inches | 9/16 | 9/16 |
| Immersion of upper tip of blade at load draught, inches | 14 9/16 | 14 9/16 |

## MAIN CONDENSERS.

As shown in Plate III, the main condensers are of the bent-tube type, the tubes being expanded into the tube sheets. The shell is of a modified oval section, narrower at the bottom, and is built in sections of galvanized steel plate 7/32-inch thick, reinforced by T-shaped stiffeners. The water ends are of No. 10 B.W.G. copper plates sweated onto cast-composition flanges and are secured to the tube sheets and to the shell by ⅝-inch Muntz metal bolts. The tube sheets are of rolled Muntz metal 1¼-inch thick, other details as follows:

| | |
|---|---|
| Maximum inside width, feet and inches | 3–05¼ |
| depth, feet and inches | 6–03 |
| Length between tube sheets, feet and inches | 14–00 |
| Tubes, number | 2,227 |
| diameter, inch | 005/8 |
| thickness, B.W.G. | 18 |
| Cooling surface, square feet | 5,100 |
| Diameter of dry air pump suction, inches | 08 |
| wet air pump suction, inches | 08 |
| circulating water inlet and outlet, inches | 19½ |
| auxiliary exhaust connection, inches | 7 |
| drain connection, inches | 3 |

Each condenser is provided with a system of baffle plates for the purpose of separating the air from the vapor and condensate as it wends its way to the wet-air pump suction. With

reference to Plate III, A and B are two solid horizontal baffle plates running the whole length of the condenser, but cut on the inboard side of their after end so as to provide a passage for the condensate. Vertical baffle plate C, which extends only a foot beyond midlength of the condenser, serves to direct the air caught between A and B through the second tube nest of the air cooler, built in the lower part of the condenser, and finally to the dry-air pump suction. Perforated baffle plate D also extends to the middle of the condenser and it is used for screening the wet-air pump suction.

Whatever condensate collects on top of baffle plate A is drawn off by a 3-inch pipe connected, through a water seal, to the wet-air pump suction.

To reduce radiation from the condensate, collecting on top of baffle A, to the air which is being cooled, the bottom of baffle A is provided with a vacuum chamber.

### MAIN AIR PUMPS.

There are two Blake vertical dual air pumps, each with one dry and one wet cylinder of 28 inches diameter, and one 12 inches diameter steam cylinder, all having an 18-inch stroke. Each wet-air pump has a 9½-inch suction from the bottom of the condenser and an 8-inch discharge to the feed filter tank. Each dry-air pump has an 8-inch suction and a 6-inch discharge to the atmosphere, with a 4-inch branch leading to a small overflow tank. When this tank is nearly filled a 3-inch float-controlled valve opens and connects the tank to the wet-pump suction, thus avoiding any loss of water through the dry-pump discharge.

### MAIN CIRCULATING PUMPS.

As shown by Plate I, there is one circulating pump to each main condenser, and it is of the single-inlet centrifugal type, driven by a single-cylinder reciprocating engine. The engine's moving parts are completely enclosed and supplied with forced lubrication by a small plunger pump driven from the engine

shaft. The pump runner and shaft is a one-piece composition casting, and is overhung from a lignum vitae bearing of liberal proportions. The pump casing is of composition, split horizontally, the lower part being strongly bolted to an extension of the engine bedplate.

*Data for one Main Circulating Pump and Engine.*

| | |
|---|---|
| Diameter of impeller, inches | 36 |
| Width of impeller at tip of blades, inches | 7¼ |
| Diameter of suction and discharge nozzles, inches | 19½ |
| and length of pump bearing, inches | 4¼ |
| steam cylinder, inches | 8 |
| piston valve, inches | 3 |
| rod, inches | 1¾ |
| Stroke of piston, inches | 8 |
| Travel of valve, inches | 2½ |
| Cut-off, mean of top and bottom | .69 |
| Diameter and length of crank-shaft bearings, inches | 3×4¼ |
| Length of connecting rod, inches | 18 |
| Diameter and length of crank-pin brass, inches | 3½×3⅛ |
| of steam pipe, inches | 2 |
| exhaust pipe, inches | 2½ |

ENGINE-ROOM AUXILIARIES.

The size and type of the various engine-room auxiliaries are shown in the appended table of pumps and pump connections; their location is shown on Plate I.

*Auxiliary Condenser and Pumps.*—The auxiliary condenser is located in the center at the after end of the auxiliary room. It is cylindrical in shape and of the bent-tube type, the tubes' ends being expanded in their tube sheets. The shell is built in three sections of galvanized Class B boiler plates 5/32-inch thick; the middle section is of 21-inch diameter, the end sections are of 19-inch diameter. The tube sheets are of rolled brass ⅞-inch thick and the water ends are composition castings designed for double flow.

There are in all 280 ⅝-inch brass tubes having a total cooling surface of 257 square feet. The mean length of tubes is about 5 feet 9 inches between tube sheets, and the tubes are supported

at two points by brass strips. Directly beneath and connected to the condenser is a Blake horizontal, simplex, combined air and circulating pump.

The auxiliary exhaust connection to the condenser has a diameter of 6-inches.

*Feed and Filter Tank.*—This is located at the forward end of the auxiliary room. It has a total capacity of 800 gallons, of which 200 gallons belongs to the filter chambers and 600 gallons to the feed tank proper. Each main air pump has an independent connection on the after side of the tank, near the top, discharging into separate filter chambers. The trap drains are led through coils at the tank's bottom and discharge through vertical pipes into each filter chamber close to the main air-pump discharge. Each filter chamber has a 5-inch by-pass pipe and gate valve to the feed tank proper, ¾-inch drain and 2-inch vapor connections. The tank has a 4-inch overflow pipe, one-half inch lime-water connection, and is fitted with a gage glass and a thermometer.

FORCED-LUBRICATION SYSTEM.

The main turbine journals and thrust bearings, as well as the main circulating and reciprocating cruising-engine journals, crank-pin and crosshead bearings, valve-stem guides, links and eccentrics, are fitted with forced lubrication.

The system includes the following:

One 200-gallon oil-drain tank located in the bilge close to the after engine-room bulkhead.

One 158-gallon settling tank, located under the deck, fitted with steam coils.

One horizontal double-flow oil cooler having a cooling surface of about 80 square feet.

Two 4½ by 6 by 6, vertical, piston, double-acting, simplex oil-pressure pumps and one pump of the same size and type for oil-cooler circulating water.

From reservoirs formed in the bearing passes the oil drains by gravity into the drain tank, where it is drawn by the oil

pump and delivered through the cooler to the bearings at a specified pressure of about 10 pounds. Macomb strainers are fitted on the suction side of the pump and a branch pipe is provided on the discharge side for by-passing the oil cooler.

The oil pumps can also, through a one-inch connection, discharge into the settling tank, which is provided with a steam coil; and from this tank the oil can be drained back to the system by gravity. Before discharging into the drain tank, returns from the main turbine journal and thrust-bearings pass through a sight-feed manifold provided with thermometers lighted by an incandescent lamp so that they can be read from the floor.

A thermometer is also provided on the discharge pipe from the cooler to show the temperature of the oil entering the bearings.

*Oil Cooler.*—The oil cooler is of the horizontal double-tube type. The oil flows through the annular space between the two tubes while the cooling water flows through the inner tubes and around the outer tubes. These are expanded into rolled brass plates bolted to the shell, the inner tubes project beyond into cast composition plates, which form part of the oil heads, and the joints are made tight by means of cotton tape packing and threaded brass ferrules. The oil enters at the top of the outboard end of the cooler, and flows out at the bottom of the same end, and the oil heads are so designed as to provide four passes of constant increasing area, the number of tubes being 5, 6, 7 and 14, respectively.

The cooling water connections are on the inboard end of the cooler, the inlet being at the bottom. The water ends are designed for two passes, but after flowing through the inner tubes, the cooling water is again directed, by means of a 2½-inch copper pipe, through the shell and around the outer tubes.

The shell is of copper about .18 inch thick, 14 inches in diameter, containing 32 double tubes with an aggregate area of 80 square feet. The inner tubes are ¾-inch external diameter, .095 inch thick and 4 feet 10½ inches long between tube

sheets; the outer tubes are 1¼-inch external diameter, .109-inch thick and 4 feet 4 inches long between tube sheets.

### OIL TANKS.

There are located in the auxiliary room the following:

(A) One 200-gallon lubricating-oil storage tank, subdivided in two compartments of equal capacity.

(B) One 50-gallon lamp oil tank.

(C) One ten-gallon cylinder oil tank.

These tanks can be filled from the deck by means of a funnel and 1¼-inch filling pipes.

Each tank has a glass gage and a brass strip, near the gage, graduated at intervals corresponding to 10 gallons.

Tank A has a 1¼-inch connection on the suction side of the forced lubrication pump for making up or filling the system.

Tanks A and B are provided with a ¾-inch lock cock and drip pan.

### WATER SERVICE.

The top and bottom halves of the main turbine bearings, the thrust horseshoes and the lower half of the line-shaft bearings are fitted for circulation of cooling water.

In addition a one-inch spray connection is provided to each main bearing and three-quarter-inch spray connection to each spring bearing. The water is supplied through a two-inch connection taken from the discharge pipe of each fire and bilge pump before they unite into the fire-main, and this main supply pipe gradually decreases in size as branches are led to the various bearings.

The drains from each bearing unite into a common pipe 2½-inch diameter and discharge overboard. Two one-inch connections are also taken from the fire main in the shaft alley for flushing out the stern tube.

### BOILERS.

Steam is supplied by four oil-burning Yarrow boilers arranged in pairs in two separate compartments. They are designed for a working pressure of 265 pounds gage, and tested

under steam at a pressure of 400 pounds. Each boiler has an independent smoke pipe of elliptical section, having an area of 18.36 square feet, the top of the smoke pipe is 31 feet 9 inches above the ash pans.

The tubes are of seamless steel obtained by the solid cold-drawn process. They are arranged symmetrically in two nests, each having 18 rows. The tubes of the two inner rows are slightly bent to provide for expansion and are of 1¼ inch external diameter; the remaining tubes are straight and have an external diameter of one inch.

The firerooms are operated under the close forced-draft system, the air pressure at full power being limited to five inches of water.

Each boiler has nine oil burners of the Yarrow-Fore River type. Each burner projects into a truncated cone of steel plate fastened to the boiler front and provided with slots controlled by louvres to regulate the admission of air. The entire boiler front is protected from radiation by an outer plate forming a deep air space, and air is admitted to the burner through non-return swing doors.

*Data for One Boiler.*

| | |
|---|---|
| Heating surface, square feet | 5,375 |
| Furnace volume, cubic feet | 630 |
| Ratio heating surface to furnace volume | 8.53 |
| Total number of one-inch tubes | 2,656 |
| 1¼-inch tubes | 218 |
| Pitch, one-inch tubes | 1½ |
| 1¼-inch tubes | 1¾ |
| Drum, upper, inside diameter, inches | 43 |
| thickness of shell, inches | $0\frac{9}{16}$ |
| thickness of tube sheet, inches | $1\frac{5}{8}$ |
| Drum, lower, inside width, inches | $27\frac{11}{16}$ |
| depth, inches | $19\frac{25}{32}$ |
| thickness of shell, inches | $0\frac{19}{32}$ |
| tube sheet, inches | 1¼ |
| Diameter of main stop valve, inches | 7 |
| safety valves (two duplex, American Steam Gauge Co.), inches | 4 |
| main feed, stop and check valves, one to each lower drum, inches | 2 |

| | | |
|---|---|---|
| Diameter of | auxiliary feed stop and check valve, inches | 3 |
| | surface-blow valve, inches | 1¼ |
| | bottom-blow valves (two Nolan, seatless, hollow piston), inches | 1½ |

Each boiler is provided with Yarrow's latest system of feed, and consists of one 2-inch main feed connection to each lower drum, through an internal feed pipe extending practically to the end of the drum, and discharging into a pocket formed by a steel plate closed at both ends and which partitions off most of the tubes in the two outer rows. The feed water is thus forced to ascend through these tubes to the upper drum at an increased velocity. Its discharge into the upper drum is directed by a short baffle plate, toward the surface of the steaming level. The object of these baffles is to afford a more effective and thorough mixture with the hotter water, and at the same time avoid the eventual condensation of the rising steam. Owing to the increased velocity of flow and the higher temperature difference, this system of feeding should increase the heat transference of the outer tubes and slightly improve the efficiency of their heating surface.

The introduction of the feed through the comparative colder tubes should also favor the liberation of any impurities contained in the water, and confine the formation of scales to this part of the heating surface where, owing to the lower temperature, it should prove less detrimental.

If these suppositions are borne out in operation, increased durability of the tubes should accrue from this system.

## FUEL-OIL SYSTEM.

This system consists of two light-service booster pumps, four heavy-service supply pumps, two oil heaters, and the oil-storage tanks in the forward and after holds with their equipment of valve manifolds, strainers and piping.

The suction pipe of each storage tank is supplied with a steam coil at its end, and each tank has connections for draining the water which has settled in the oil tanks and discharge it overboard.

Each group of tanks is controlled by a manifold located on the extreme bulkhead limiting the machinery spaces. These two manifolds are connected by a common suction main with branch connections to the booster pumps and to deck connections for taking on oil, so that either pump draws from or discharges to all storage tanks and deck connections.

The service pumps can draw oil directly from the storage tanks or from suction main via booster pumps' discharges, and deliver it to the burners through the oil heaters or by-passes.

A 2½-inch Macomb strainer is fitted on the suction side of each supply pump and a double 2-inch filter is fitted on the discharge between the pump and the oil heater.

The pipe supplying the burners of each boiler is provided with a master cut-out valve controlled from both the firerooms and deck.

Each boiler has nine burners of the mechanical atomizing type of Yarrow-Fore River design. The tip has a bore of .085-inch diameter and is controlled by a needle valve for regulating the spray. Each burner has a cut-out cock and a small basket strainer. Smaller bore burners are provided for starting when no steam or air is available, and for this purpose a small two-cylinder Rumsey hand pump is fitted in each fireroom of a sufficient capacity to deliver oil to two burners at about 200 pounds pressure.

All valves and fittings are of composition. All pipes are seamless-drawn steel expanded into Class B forged-steel flanges and tested to the following pressures: Booster-pump suction, 50 pounds; discharge, 100 pounds; supply-pump discharge, 600 pounds; working pressure, 300 pounds.

### FIREROOM AUXILIARIES.

*Forced-Draft Blowers.*—There are two 39-inch outside-diameter, single-inlet Sturtevant cone fans in each fireroom, mounted on vertical shafts in the base of the fireroom ventilators. These ventilators have an inside diameter of 40 inches, and a light plate in the shape of a truncated cone leads the

air directly to the inlet of the fan which has a diameter of 30½ inches.

Each blower is provided with diffuser plates of 52 inches diameter, spaced 5½ inches apart at their periphery. Each blower is rated at 23,000 cubic feet of free air per minute against a static pressure of 5 inches of water, and when running at 1,400 r.p.m. requires about 61 shaft horsepower. Each blower is driven by a vertical Curtis steam turbine, having a single wheel of 24 inches mean diameter with five rows of moving blades. Steam is supplied through a 2-inch double-beat valve controlled by an adjustable governor driven by a worm gear on the shaft and geared down in the ratio of five to one. Speed regulation is also afforded by valve-controlled steam nozzles designed to expand from 250 to 10 pounds gage pressure. The exhaust pipe is 3½ inches diameter. The weight of the moving parts is taken up by a ball-thrust bearing on the lower end of the shaft. The thrust and vertical bearings run partially flooded in an oil bath, the oil being pumped from the base to the top of the bearings through spiral grooves cut in the brasses. Lubrication to the governor parts is by a sight-feed box.

### FEED-WATER HEATERS.

There are two feed-water heaters, one in each fireroom, of the Reilly multicoil vertical type. The shell is of steel plate with bumped heads, the internal diameter is 78 inches, the height from the base to the face of flange for the auxiliary exhaust connection at the top is 4 feet 10½ inches. The shell is provided with a large inspection door. The heating agent is the auxiliary exhaust steam which enters at the top through a 4½-inch connection and leaves at the bottom through a 1½-inch drain.

The coils are secured to ring-shaped composition headers, each carrying a four-inch feed connection, the bottom one being the feed inlet. There are 19 coils with a total heating surface of about 80 square feet. Each coil is made up of 19 turns of one-inch copper tubing, .065 inch thick, the ends be-

ing belled out and secured to the headers by means of screwed composition coned unions. The coils have an outer diameter of 4⅝ inches and are 32⅝ inches long.

For other fireroom auxiliaries see table of pumps and pump connections.

### MAIN STEAM PIPING.

Each boiler has a 7-inch steam connection, the two forward boilers combine into one 9½-inch main line on the starboard side, with an expansion joint on each bulkhead. The two after boilers combine into one 9½-inch main line on the port side with an expansion joint forward of the engine-room bulkhead. Each main line has an angle bulkhead stop valve, on engine-room side, with a 6½-inch cross connection. Outboard of each bulkhead stop are the 9½-inch astern and ahead turbine throttles and a 3½-inch connection to the cruising engine, their position being in the order named.

### AUXILIARY STEAM PIPING.

There is a 3½-inch stop valve on each boiler for supplying the auxiliary steam line. In the forward fireroom branches for the various auxiliaries are taken off these connections and the balance combines into a 3½-inch pipe which is led on the starboard side to the after fireroom, and from here, after supplying the after fireroom auxiliaries, one 4-inch branch on each side of the ship is carried aft through the engine room to the auxiliary room. These two branches are cross-connected, a first time in the after boiler room and again forward of the after engine room bulkhead, where their size is reduced to 3½ inches. From this line connections are made to the various auxiliaries and branches led forward and aft of the machinery spaces to the deck machinery, heating system and galley.

### AUXILIARY EXHAUST PIPING.

Beginning in the forward fireroom with a 2½-inch connection from the deck machinery, the auxiliary exhaust main runs on the port side, collecting the exhaust steam from the

various fireroom and engine-room auxiliaries, gradually increasing to 8 inches in diameter in the after fireroom. From here it is led aft through the engine room to the auxiliary room, where it runs athwartship and, crossing to the starboard side, is led again back to the engine room, where it is reduced to a diameter of 6½ inches, terminating with branches to the first and 18th stage of both turbines. The following are the most important connections to the auxiliary exhaust main:

2½ inches from forward deck machinery;

3½ inches from each blower turbine;

4½ inches to each feed heater;

6½ inches to 1st and 18th stage of each main turbine;

7 inches to each main condenser through a spring relief valve;

6 inches to auxiliary condenser through a spring relief valve.

The 4-inch exhaust from each of the dynamo turbines combine into a 6-inch pipe with connections to the auxiliary exhaust main or through spring-relief valves to the auxiliary condenser or to the port main condenser.

Judging from the size of the feed-water heaters and the various connections, it is seen that the *Duncan's* auxiliary exhaust piping differs from other vessels of her class, as most of her auxiliary exhaust is utilized through the main turbines.

### MAIN AND AUXILIARY FEED SYSTEM.

An 8-inch suction main begins aft at the feed and filter tank, with a 5½-inch connection to the channelway cross-connecting the main air pumps, and is led forward to the main feed pumps and to the after fireroom. Here it divides into two 5½-inch branches, one to each auxiliary feed pump in the two firerooms, and connects through 5½-inch pipes to the port and starboard reserve feed tanks in the after fireroom. A 1½-inch make-up feed line draws from the cofferdam and from each reserve feed tank and connects to the suction of each auxiliary feed pump and to the pipe cross-connecting the suc-

tions of the main air pumps. A 6-inch discharge main collects the two 4-inch branches from the main feed pumps and continues forward with a 4-inch branch to each heater or by-pass after which it branches to the lower drums of each boiler.

Each auxiliary feed pump discharges directly through the auxiliary feed line into the upper drum of each boiler of its own fireroom, or via the heaters or by-passes into the main feed line to the boilers of either fireroom.

### EVAPORATING AND DISTILLING PLANT.

Plate I shows the evaporators to be located in the auxiliary room with the distillers in the forward part of the after deck house at a high elevation. There are two high-pressure, single-effect evaporators of a combined nominal capacity of 3,750 gallons of water per 24 hours and two distillers of a combined nominal capacity of 2,500 gallons of potable water per 24 hours.

The evaporators take steam from the auxiliary steam line and drain through traps to the feed tank and main condensers.

The evaporator coils are supplied by the evaporator feed pump, which draws from the sea or from the discharge of the distiller circulating water and discharges through or by-passes a small feed heater, using vapor from the evaporators as a heating medium. From the coils the vapor is directed to the distillers or into the auxiliary exhaust line. Each distiller drains by gravity to a 20-gallon reservoir tank to be directed by the fresh-water pump to the reserve feed tanks, ship's fresh-water tanks and cofferdam. The circulating water for distillers is taken off the discharge of either fire and bilge pump, and from the distillers passes directly overboard.

*Data for One Evaporator.*

| | |
|---|---|
| Type | Vertical, Reilly, Multicoil. |
| Diameter, inside, inches | 32 |
| Height over all, feet and inches | 6–06 |
| Number of coils | 10 |
| Diameter of coil tubes, inches | 1 |

| | |
|---|---|
| Thickness of coil tubes, B.W.G | 16 |
| Heating surface, square feet | 61.5 |
| Diameter of steam nozzle, inches | 1½ |
| vapor nozzle, inches | 2½ |
| feed valve, inches | 1 |
| blow valve, inches | 1¼ |

*Data for One Distiller.*

| | |
|---|---|
| Type | Vertical, Reilly, Multicoil. |
| Diameter, inside, inches | 13 |
| Height between covers, feet and inches | 5–5⅝ |
| Number of coils | 3 |
| Diameter of coil tubes, inches | 1 |
| Thickness of coil tubes, B.W.G | 16 |
| Cooling surface, square feet | 18.5 |
| Diameter of vapor inlet, inches | 1½ |
| drain connection, inch | 0¾ |
| circulating water connections, inches | 1½ |

TRAPS AND DRAINS.

The following table gives a list and size of drains connection to automatic traps.

| Number. | Size, inches. | Service and location. | Working pressure, pounds gage. | Size of— | |
|---|---|---|---|---|---|
| | | | | Inlet, inches. | Outlet, inches. |
| 2 | ½ | Main steam pipe, engine room side of bulkhead. Stop valves, port and starboard. | 265 | ½ flanged. | ½ screwed. |
| 2 | ½ | Auxiliary steam separator, engine room, port and starboard. | 265 | ½ flanged. | ½ screwed. |
| 1 | ½ | Auxiliary exhaust drain, after fireroom, port side. | 50 | ½ screwed | ½ screwed. |
| 2 | ¾ | Feed-water heaters | 50 | 1 flanged. | 1 screwed. |
| 2 | ¾ | Fuel-oil heaters | 265 | 1 flanged. | 1 screwed. |
| 2 | ½ | Evaporator steam heads | 265 | ½ flanged. | ½ screwed. |
| 1 | ½ | Galley steam drains | 30 | ½ screwed | ½ screwed. |
| 1 | ½ | Drains from forward heating system. | 50 | 1 screwed | 1 screwed. |
| 1 | ½ | Drains from after heating system | 50 | ½ screwed | ½ screwed. |
| 1 | ½ | Whistle | 265 | ½ flanged. | ½ screwed. |

All traps are of the ball-float, positive-action type. They are supplied with a glass gage and with by-pass connections to the drain main.

Drains from throttle valves are led direct to the drain main between the traps and condensers. Two main drain pipes run the whole length of the machinery spaces, one pipe on each side of the ship, and collect all various drains, gradually increasing in size to 1½ inches diameter, and they have full-size connection to each main condenser, to feed and filter tank and to auxiliary condenser.

Each turbine has an independent draining system; connections from the bottom of the various stage chambers lead through a cut-out valve, whose stem has a hand-wheel extension above the floor, to a 1½-inch main pipe, and by means of a steam ejector the water is lifted directly to the turbine's own condenser. The discharge is at the top of the condenser's connection from the traps' main with a cut-out and a non-return swing-check valve.

## MACHINERY WEIGHTS.

The following table gives the actual machinery weights by groups as furnished by the contractors:

MACHINERY WEIGHTS FOR U. S. S. DUNCAN.

| Name of Group. | Weight, Pounds. |
|---|---|
| Main turbines | 124,318 |
| Operating valves and pipes | 11,969 |
| Turning gear | 776.5 |
| Oiling gear and clearance indicators | 1,557.5 |
| Line, stern tube and propeller shafts | 34,346.5 |
| Shaft bearings | 6,647 |
| Propellers | 6,182 |
| Main air pumps | 13,868.5 |
| Main condensers | 38,141 |
| Main circulating pumps and engines | 7,411.5 |
| Miscellaneous pumps | 19,880 |
| Stacks and uptakes | 20,457 |
| Boilers | 211,517.5 |
| Forced-draft blowers | 6,652 |

| | |
|---|---|
| Clothing and lagging | 17,291 |
| Gages, instruments, etc. | 1,454 |
| Tools, spare parts and misc. outfit (carried on board) | 13,768 |
| Engineering piping | 94,776 |
| Floors, gratings, ladders, handrails, etc. | 15,305.5 |
| Feed, oil and waste tanks | 5,694 |
| Auxiliary condenser and feed heaters | 4,915 |
| Heating system | 3,577 |
| Evaporators and distillers | 3,221 |
| Fuel oil burning apparatus | 3,803 |
| Clutch gear | 1,462 |
| Cruising engines complete | 29,439 |
| Total weight, dry | 698,429.5 |
| Water in condensers | 10,488 |
| feed tank | 4,426 |
| stern tube | 902 |
| boilers | 38,248 |
| auxiliary pumps, pipes, etc. | 11,713 |
| distilling plant | 2,078 |
| Total weight water | 67,855 |
| Total weight, wet | 766,284.5 |
| Total weight of spares sent to Navy Yard | 2,257 |

TRIALS.

The following trials were required:

(*a*) A progressive trial over the measured-mile course at Rockland, Maine, to ascertain the revolutions of the propellers corresponding to the measured speeds, extending from maximum speed down to a speed of twelve knots, about twenty-six runs to be made over the course in order to adequately cover the range of speed desired.

(*b*) A full-speed trial of four-hours' duration in the open sea in deep water, at the highest speed attainable. During this trial the steam pressure at the turbine was not to exceed 240 pounds gage. Guaranteed fuel-oil consumption, 686 pounds per knot run.

(*c*) A fuel-oil and water-consumption trial of four hours in the open sea, in deep water, at an average uniform speed of twenty-four knots, with all necessary auxiliaries in opera-

tion, but not including the evaporating and distilling plant. Guaranteed fuel-oil consumption 402 pounds per knot run.

(*d*) A four-hours' fuel-oil and water-consumption trial in the open sea at an average uniform speed of fifteen and one-half knots. This trial to be made as nearly as possible under service cruising condition with cruising engines connected and in use. Guaranteed fuel-oil consumption 141 pounds per knot run.

(*e*) An endurance trial of twenty hours in the open sea at an average uniform speed of fifteen and one-half knots, following as close as possible the trial prescribed in clause *d*, and to be made under the same conditions as on that trial. Measurements of fuel-oil and water consumptions were not required.

(*f*) A fuel-oil and water-consumption trial of four hours in the open sea with the cruising engines connected and in use, at an average uniform speed of twelve knots. Guaranteed fuel-oil consumption 110 pounds per knot run.

In addition to these trials backing and turning trials were specified.

STANDARDIZATION TRIALS.

These trials began at 6:33 A. M., and they were completed at 11:52 A. M. The conditions were very good; the sky overcast and hazy, and the sea smooth and dead calm.

The draught was carefully taken by the internal draught gages, with the anchor up and the ship dead in the water, both at the beginning and at the end of trials.

| | Beginning of Trials. | End of Trials. |
|---|---|---|
| Draught, forward, feet and inches.. | 8–10 1/8 | 9–07 3/8 |
| aft, feet and inches..... | 10–04 7/8 | 9–04 1/4 |
| mean, feet and inches..... | 9–07 1/2 | 9–05 13/16 |
| Trim, inches..... | 18 1/8 by stern | 03 1/8 by head |
| Displacement—corrected for trim and density, tons..... | 1,077 | 1,049.2 |

It was estimated that the displacement at the middle of the five high-speed runs on the standardization trials was 1,055.5 tons and the mean draught 9 feet 6¼ inches, even keel.

Throughout the trials the reciprocating engines were uncoupled and not in use.

The results of the standardization trials are contained in Table III, from which the curves of Plate V were constructed. The following mean revolutions of the two shafts were required for the various contract speeds:

| | | |
|---|---|---|
| 12 | knots | 194.0 |
| 15½ | knots | 251.3 |
| 24 | knots | 410.5 |
| 29 | knots | 566.1 |

FOUR-HOUR FULL-SPEED TRIAL.

This trial began at 3:30 P. M., July 5th, and was completed at 7:30 P. M. the same day. The weather conditions were excellent; it was overcast and slightly hazy, the sea was smooth, and no wind blowing.

During this trial the auxiliary exhaust was used on feed heaters and the surplus led to the 18th stage of the main turbines. All trap drains were led to the starboard main condenser. The feed-water measurement includes the exhaust from all auxiliaries in operation and all drains. The amount of reserve feed used during the trial was 1,019 gallons or 8,476 pounds. Auxiliaries in engine and firerooms were operated as shown by Tables IV and V, with fire and bilge pump on sanitary system.

FOUR-HOUR 24-KNOT TRIAL.

This trial began at 7:00 A. M., July 7th, and was completed at 11:00 A. M., the same day. The weather conditions were excellent; it was generally clear and pleasant; wind southwest; force two to three, and a heavy, moderate southwesterly swell.

During this trial the auxiliary exhaust was led to the 18th stage of main turbines; feed heaters not in use. All trap drains were led to the starboard main condenser. The feed-water measurement includes the exhaust from all auxiliaries in operation and all drains. The amount of reserve feed used during the trial was 995 gallons, or 8,284 pounds. Auxiliaries were operated as shown by Tables IV and V, with fire and bilge pump on sanitary system.

FOUR-HOUR 12-KNOT TRIAL.

This trial began at 2:50 P. M., July 7th, and ended at 7:50 P. M. of the same day. The same weather conditions of the morning prevailed on this trial. The cruising engines were coupled up and in use, as required by the contract.

During this trial the auxiliary exhaust was led to the first stage of the main turbines; feed heaters not in use. All trap drains were led to the starboard main condenser. The feed-water measurement includes the exhaust from all auxiliaries in operation and all drains. The amount of reserve feed water used during the trial was 232.5 gallons, or 1,940 pounds. Auxiliaries were operated as shown in Tables IV and V, with fire and bilge pumps on sanitary system.

FOUR-HOUR 15½-KNOT TRIAL.

This trial began at 6:15 A. M., July 8th, and ended at 10:15 A. M. of the same day. The weather conditions were excellent, the wind was west, force 2, and the sea smooth. The cruising engines were coupled up and in use, as required by the contract.

During this trial the auxiliary exhaust was used on first stage of the main turbines; feed heaters not in use. All trap drains were led to starboard main condenser. The feed-water measurement includes the exhaust from all auxiliaries in operation and all drains. The amount of reserve feed water used during the trial was 215 gallons, or 1,792 pounds. Auxiliaries

were operated as shown in Tables IV and V, with fire and bilge pump on sanitary system.

### 15½-KNOT ENDURANCE TRIAL.

According to the contract this trial followed shortly after the four-hour 15½-knot steam and fuel-consumption trial, but it had hardly begun when it had to be discontinued, owing to the failure of the piston rings in both H.P. cylinders.

After new rings had been installed at the works of the contractors the *Duncan* resumed the trial at 11:00 P. M., July 9th, and was completed at 8:00 P. M., July 10th. The weather conditions were, on the whole, excellent. At the commencement of the trial the wind was southwest, with a moderate sea. The wind fell toward morning, and the latter half of the trial was run with light variable winds and a smooth sea.

The average r.p.m. for 21 hours was 248.94, corresponding to a speed of 15.3 knots. Although no fuel-oil consumption measurements were required in this trial, some observations were made by the contractor to determine the relative economy with feed water entering the water drums through main feed checks, and with feed entering steam drum through auxiliary check, all other conditions being maintained constant. The elapsed time for the consumption of three measuring tanks full of oil was found to be 1 hour, 35 minutes, 12 seconds for the first case, and 1 hour, 32 minutes, 30 seconds for the second case. These results show a fuel economy of about 6 per cent. in favor of the main-feed system.

After the completion of the endurance run at 15½ knots, the water and oil consumptions were obtained with cruising engines uncoupled, making 15½ knots under turbines alone. Four nozzles were opened on each turbine; pressure in steam chests, 235 pounds gage; vacuum at condenser, 28 inches.

The following observations and deduced results show the relative economy of the combination of reciprocating cruising engines and turbines over the main turbines only:

| | Cruising engines and turbines. | Turbines alone. |
|---|---|---|
| Revolutions per minute | 249.81 | 249.21 |
| Corresponding speed from standardization curve | 15.40 | 15.37 |
| S.H.P. from standardization curve | 1,280 | 1,265 |
| Steam consumption, all purposes, pounds per hour | 25,880 | 48,540 |
| Steam consumption, all purposes, pounds per S.H.P. | 20.2 | 38.35 |
| Steam consumption, all purposes, pounds per knot run | 1,680 | 3,160 |
| Fuel-oil consumption, pounds per hour | 2,025 | 3,756 |
| pounds per knot run | 131.5 | 244.3 |

## TURNING AND BACKING TRIALS.

After the standardization trials had been completed the vessel was backed over the measured mile in order to determine the number of revolutions to give the vessel a speed of about 14 knots when going astern. Two runs were made over the course while going astern, the main turbines only being used. The following results were obtained:

| Number of Run. | Revolutions per minute. Starboard. | Port. | Average. | Speed in knots. |
|---|---|---|---|---|
| 1 | 293.34 | 296.22 | 294.78 | 13.91 |
| 2 | 310.16 | 308.29 | 309.23 | 13.942 |
| Average | | | 302.00 | 13.926 |

After the completion of the four-hour 24-knot trial the speed of vessel was increased to 29 knots and the steering gear was tested, the ship being turned through two complete circles, first with full left rudder, then with full right rudder. The following observations were made:

| | Left rudder. | Right rudder. |
|---|---|---|
| Time to shift wheel at bridge, seconds | 8½ | 14 |
| Time at rudder head, seconds | 8 | 14 |
| Time to swing 360 degrees, minutes and seconds | 3:40 | 3:45 |
| Estimated diameter of turning circle, yards | 600 | 600 |
| Initial angle of heel, degrees | 3 | 3 |
| Final angle of heel, degrees | 2 | 2 |

This completed the steering tests with the vessel going ahead.

The ship was then brought onto a straight course again and the backing trial was made. With the vessel making approximately 29 knots the engine telegraphs were thrown "full speed astern," and after the vessel had come to a dead stop the order "full speed ahead" was given. The following engine-room observations were taken from time signal to reverse was received:

| | Starboard. | Port. |
|---|---|---|
| Turbines began turning over astern, seconds | 51 | 40 |
| Turbines running full speed astern, minutes and seconds | 1:32 | 1:10 |
| Bridge time from signal to dead in the water, minutes and seconds | 1:20 | |
| Distance "head reached," yards | 325 | |

From the signal for the second reversal the following times were noted:

| | Starboard. | Port. |
|---|---|---|
| Began turning over ahead, seconds | 15 | 15 |
| Full speed ahead, seconds | 30 | 30 |

At the completion of the endurance trial at 15¼ knots with reciprocating cruising engines, further backing trials were made to test the backing qualities of the reciprocating engines. After steam had been raised in a second boiler, both turbines were reversed, cruising engines remaining coupled with their valve gear thrown full astern, but no steam was turned into them. Under these conditions they revolved easily and without any pounding. The maximum number of revolutions reached was 272 per minute. On steaming ahead again at 15½ knots with steam entering cruising engines only, signal was given for full speed astern, steam being admitted to cruising engines only.

The following engine-room observations were noted from time signal was received:

| | Starboard. | Port. |
|---|---|---|
| Engines began to back, seconds | 33 | 48 |
| Engines backing full speed (about 220 r.p.m.), minutes and seconds | 2:33 | 2:20 |

Signal was then made for full speed ahead, meaning 15½ knots, with steam to cruising engines only, with results as follows:

| | Starboard. | Port. |
|---|---|---|
| Engines began turning ahead, seconds | 45 | 35 |
| Engines running full speed ahead, minutes and seconds | 1:00 | 0:45 |

This completed the backing trials.

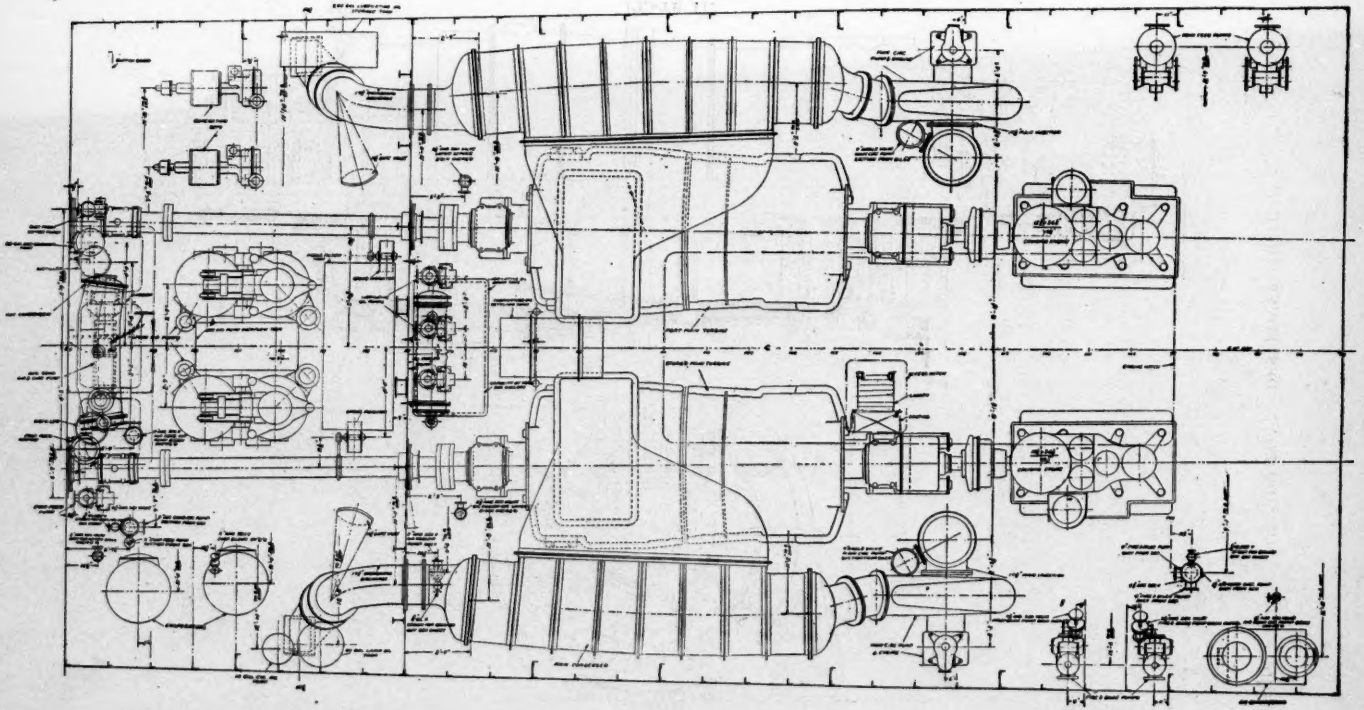


PLATE I.

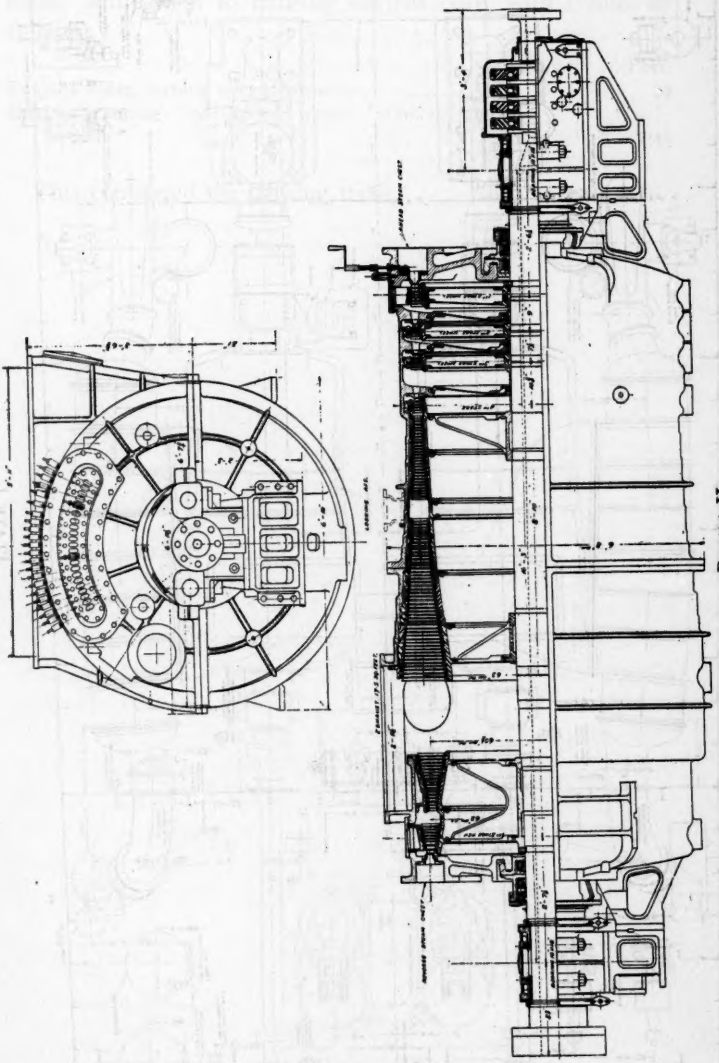


PLATE II.

PLATE II.

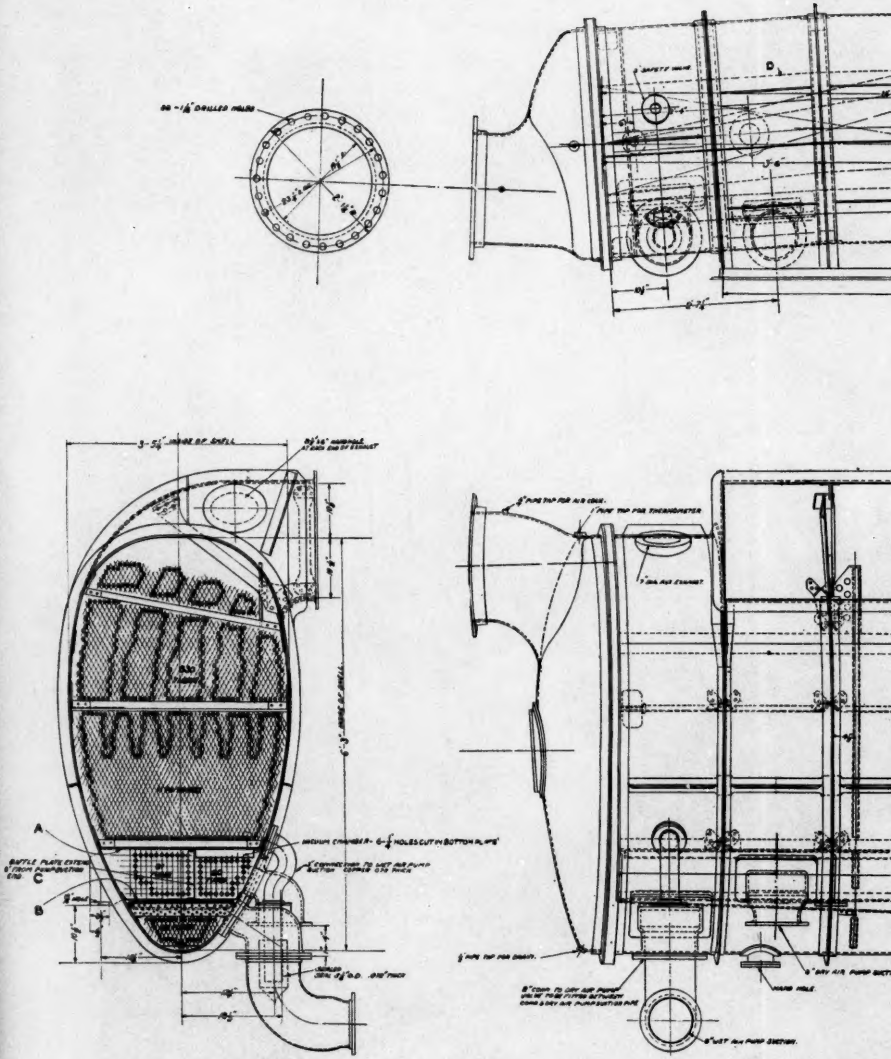
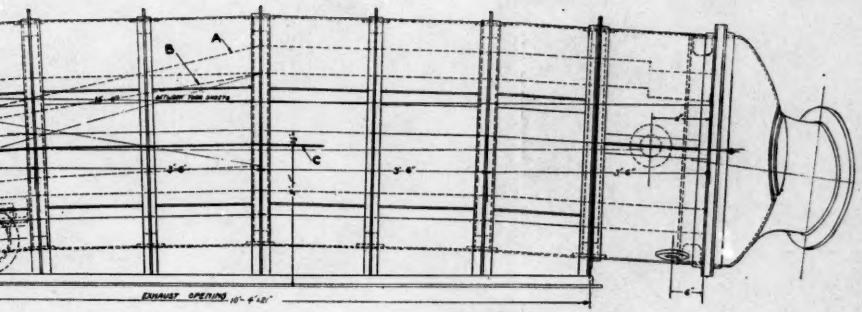


PLATE III.



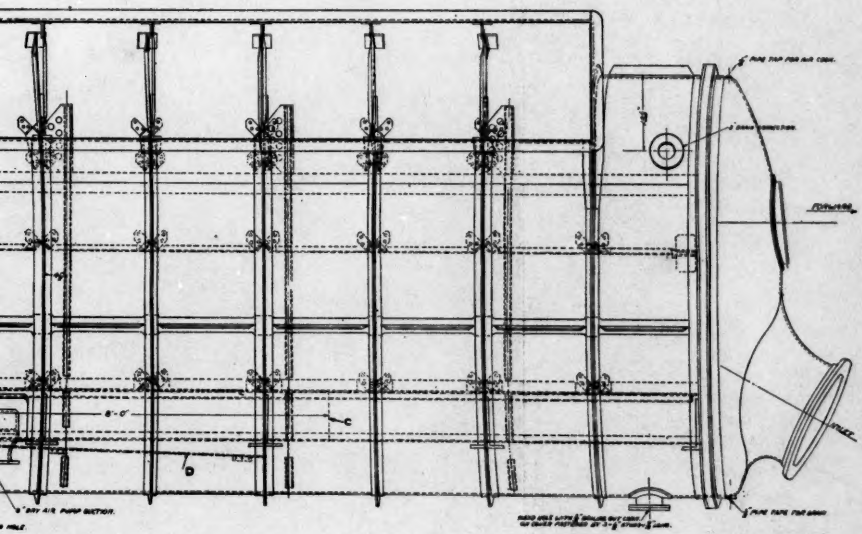




PLATE III.

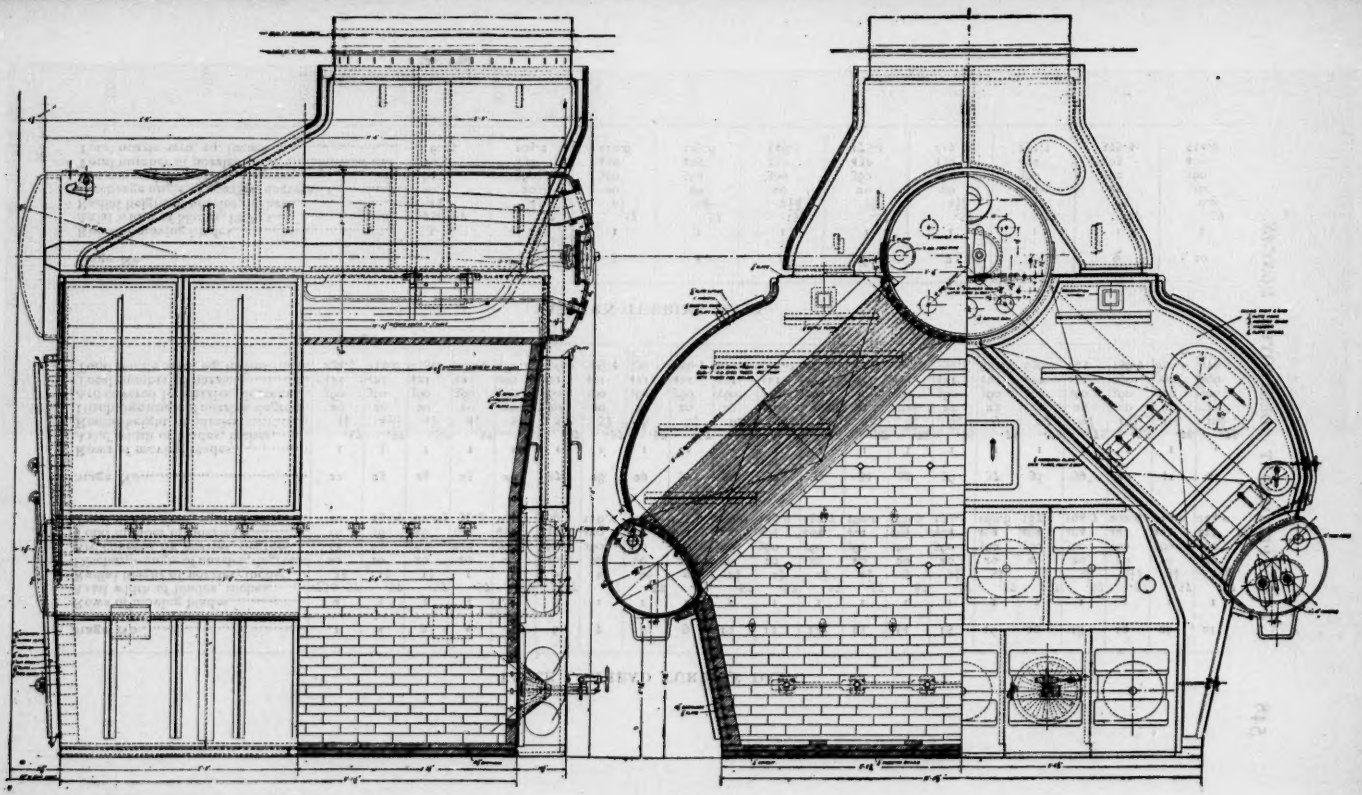


PLATE IV.

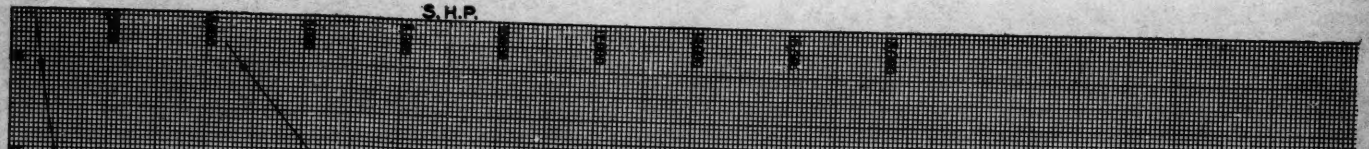
TABLE I.—AHEAD TURBINE DATA.

| Stage No. | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 | 17 | 18 | 19 | 20 | 21 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Rows of moving blades | 4 | 3 | 3 | 3 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 |
| Axial width of blades, inches | .75&.57 | .57 | .57 | .57 | .57 | .57 | .57 | .57 | .57 | .57 | .57 | .57 | .57 | .57 | .57 | .57 | .57 | .57 | .57 | .57 | .57 |
| Radial height of nozzles, inches | 1⅛ | 1⅛ | 1⅛ | 1 | 1⅛ | 2 | 2⅛ | 2⅛ | 2⅛ | 2⅛ | 2⅛ | 2⅛ | 2⅛ | 3 | 3⅛ | 3⅛ | 3⅛ | 3⅛ | 3⅛ | 3⅛ | 4⅛ |
| Discharge angle of nozzles, degrees | 20 | 20 | 20 | 20 | 20 | 20 | 20 | 20 | 20 | 20 | 20 | 20 | 20 | 20 | 20 | 20 | 20 | 20 | 20 | 20 | 20 |
| Arc covered by nozzles, degrees | 42 | 83 | 123 | 220 | 360 | 360 | 360 | 360 | 360 | 360 | 360 | 360 | 360 | 360 | 360 | 360 | 360 | 360 | 360 | 360 | 360 |
| Total number of nozzles | 16 | 38 | 56 | 100 | 440 | 440 | 421 | 404 | 404 | 404 | 404 | 404 | 404 | 404 | 404 | 404 | 404 | 404 | 421 | 421 | 421 |
| Total nozzle area, sq. inches | 10 | 15.09 | 22.23 | 35.3 | 101.7 | 109.2 | 116.2 | 122.8 | 130.9 | 139.0 | 145.9 | 152.7 | 160.0 | 166.9 | 173.7 | 180.6 | 187.5 | 194.7 | 202.9 | 213.0 | 224.0 |

| Stage No. | 22 | 23 | 24 | 25 | 26 | 27 | 28 | 29 | 30 | 31 | 32 | 33 | 34 | 35 | 36 | 37 | 38 | 39 | 40 | 41 | 42 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Rows of moving blades | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 |
| Axial width of blades, inches | .57 | .57 | .57 | .57 | .57 | .57 | .57 | .57 | .57 | .57 | .57 | .57 | .57 | .75 | .75 | .75 | .75 | .75 | .75 | .75 | .75 |
| Radial height of nozzles | 4⅛ | 4⅛ | 4⅛ | 4⅛ | 5⅛ | 5⅛ | 5⅛ | 6 | 6⅛ | 6⅛ | 7⅛ | 7⅛ | 7⅛ | 8⅛ | 8⅛ | 9 | 9⅛ | 9⅛ | 10⅛ | 10⅛ | 10⅛ |
| Discharge angle of nozzles, degrees | 20 | 20 | 20 | 20 | 20 | 20 | 20 | 20 | 20 | 20 | 20 | 20½ | 21 | 21½ | 22 | 23 | 24 | 25 | 26 | 28 | 31 |
| Arc covered by nozzles, degrees | 360 | 360 | 360 | 360 | 360 | 360 | 360 | 360 | 360 | 360 | 360 | 360 | 360 | 360 | 360 | 360 | 360 | 360 | 360 | 360 | 360 |
| Total number of nozzles | 421 | 421 | 421 | 421 | 421 | 421 | 421 | 421 | 421 | 421 | 421 | 421 | 421 | 421 | 421 | 421 | 421 | 421 | 396 | 396 | 396 |
| Total nozzle area, sq. inches | 234.5 | 244.6 | 255.1 | 268.2 | 282.1 | 296.0 | 309.4 | 330.1 | 350.7 | 371.3 | 392.0 | 425.2 | 458.9 | 494.3 | 530.5 | 581.8 | 636.1 | 695 | 758.7 | 851.0 | 979.7 |

ASTERN TURBINE DATA.

| Stage No. | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 |
|---|---|---|---|---|---|---|---|---|---|---|
| Rows of moving blades | 5 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 |
| Axial width of blades, inches | .75&.57 | .57 | .57 | .57 | .57 | .57 | .57 | .57 | .57 | .57 |
| Radial height of nozzles, inches | 1⅛ | 2 | 2⅛ | 2⅛ | 2⅛ | 3⅛ | 4⅛ | 5 | 5⅛ | 6.2 |
| Discharge angle of nozzles, degrees | 20 | 20 | 20 | 20 | 20 | 20 | 20 | 22 | 25 | 30 |
| Arc covered by nozzles, degrees | 64 | 360 | 360 | 360 | 360 | 360 | 360 | 360 | 360 | 360 |
| Total number of nozzles | 10 | 432 | 432 | 405 | 432 | 432 | 432 | 432 | 405 | 405 |
| Total nozzle area, sq. inches | 6.75 | 104.1 | 110.6 | 125.6 | 146.5 | 175.4 | 217.7 | 291.1 | 375.4 | 514.8 |



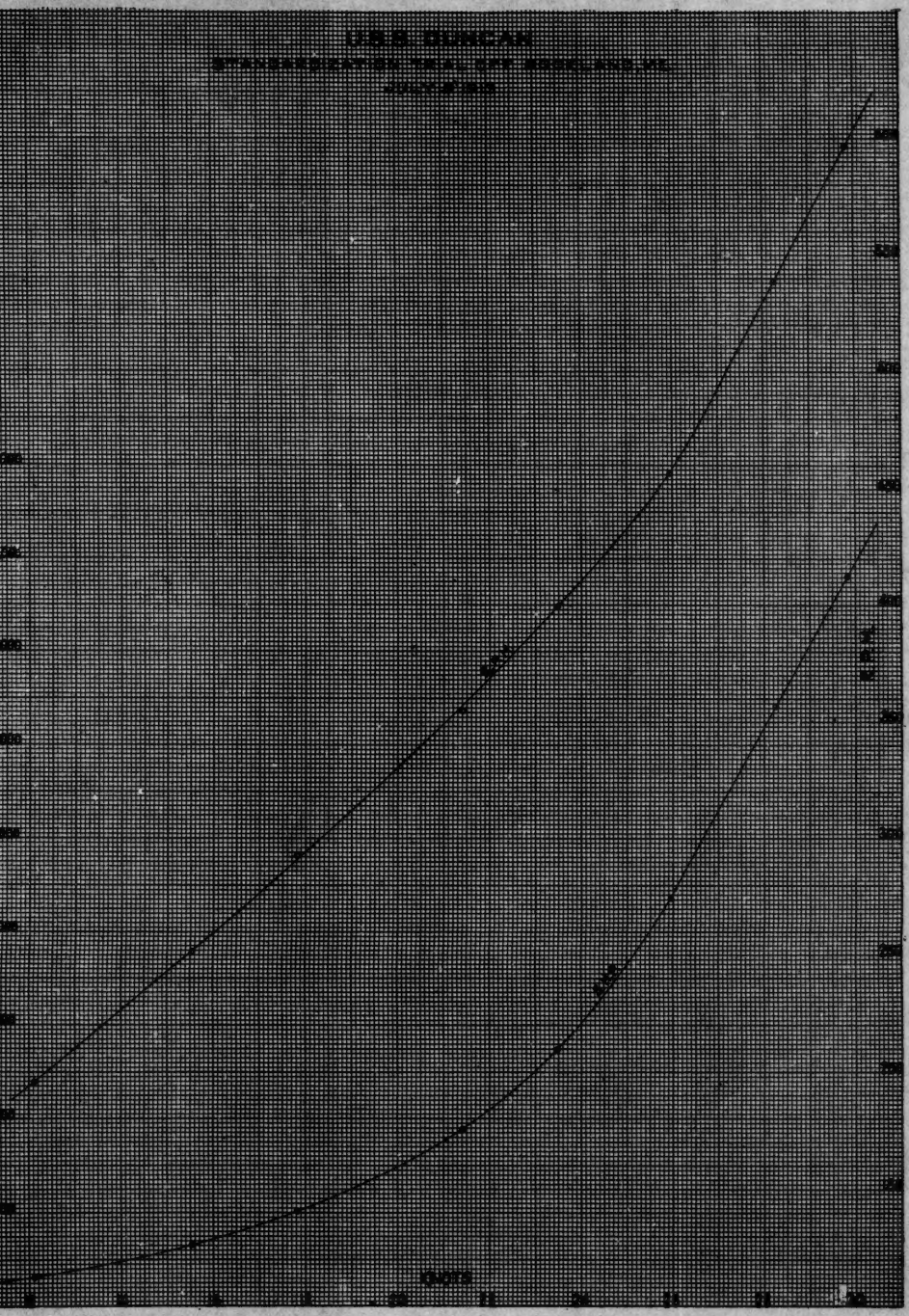


PLATE V.

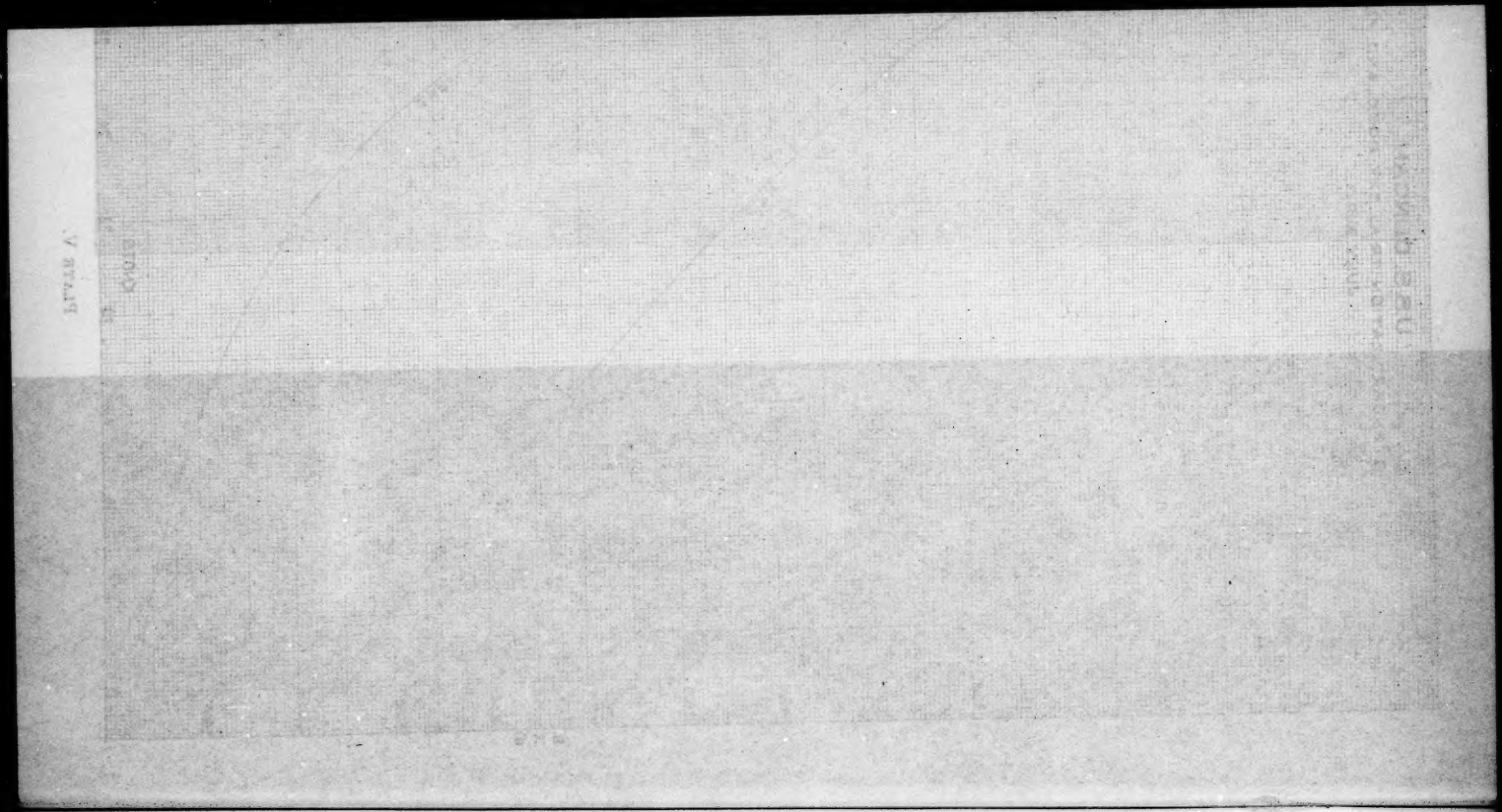


TABLE II.—PUMPS AND PUMP CONNECTIONS.

## TABLE II.—PUMPS AND PUMP CONNECTIONS.

| Number. | Name. | Cylinders. | | | | | Type. | Connections. | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Steam. | | Water. | | | | Suction. | | Discharge. | | Location. |
| | | No. | Diam. | No. | Diam. | Stroke, inches. | | Inches. | From. | Inches. | To— | |
| 2 | Main air | 1 | 12 | 1 wet<br>1 dry | 28 | 18 | Blake, vertical, twin, beam, bucket | 1½<br>8<br>8 | Reserved feed tanks.<br>Condensers, wet<br>Condensers, dry | 8<br>6 | Feed tank<br>Atmosphere. | Aux. room. |
| 2 | Main circulating | 1 | 8 | 36″ run'r. | | 8 | Engine-driven centrifugal | 19½<br>7 | Sea<br>Bilge. | 19½ | Condenser | Eng. room. |
| 2 | Main feed | 1 | 15 | 1 | 10 | 16 | Blake, vertical, piston, double-acting simplex. | 5½<br>5½ | Feed and filter tank<br>Wet-air pump cross connect.<br>Reserve feed tanks | 4<br>1½ | Main-feed discharge<br>Reserve feed tank. | Eng. room. |
| 2 | Auxiliary feed | 1 | 15 | 1 | 10 | 16 | ....do.... | 5½<br>1½ | Feed suction pipe, reserve feed tanks and cofferdam.<br>Hose connection | 4<br>1½ | Auxiliary feed<br>Hose connection. | One each fireroom. |
| 2 | Fire and bilge | 1 | 7 | 1 | 7 | 12 | ....do.... | 4<br>4<br>1½ | Sea<br>Drainage<br>Hose connection | 2½<br>3½<br>2<br>2<br>1½ | Fire main flushing system through fire main.<br>Overboard.<br>Distiller.<br>Water service.<br>Hose connection. | Aux. room. |
| 1 | Auxiliary air and circulating. | 1 | 6 | 1 air.<br>1 circ. | 8<br>8 | 7 | Blake, horizontal, combined | 4<br>4 | Auxiliary condenser<br>Sea | 3<br>4 | Feed tanks.<br>Auxiliary condenser. | Aux. room. |
| 1 | Evaporator feed | 1 | 4½ | 1 | 6 | 6 | Blake, vertical, piston, double-acting simplex. | 2<br>1½ | Sea<br>Distiller circulating water discharge. | 1½ | Evaporators | Do. |
| 1 | Dist. fresh water | 1 | 3½ | 1 | 4 | 4 | ....do.... | 1½ | Distiller reservoir tank | 1½<br>1½<br>1½ | Reserve feed tanks.<br>Fresh-water tanks.<br>Cofferdam. | Do. |
| 1 | Oil-cooler, circ'g | 1 | 4½ | 1 | 6 | 6 | ....do.... | 3 | Sea | 2½ | Oil cooler | Do. |
| 2 | Lubricating oil | 1 | 4½ | 1 | 6 | 6 | ....do.... | 3 | Lubricating-oil tanks | 2½ | Bearings through cooler | Eng. room. |
| 4 | Fuel-oil service | 2 | 6 | 2 | 3½ | 8 | Blake, vertical, piston, double-acting, duplex. | 2½<br>2½ | Oil-storage tanks<br>Booster-pump discharge. | 2 | Burners | Two each fireroom. |
| 2 | Fuel-oil booster | 1 | 4½ | 1 | 6 | 6 | Blake, vertical, piston, double-acting, simplex, light service. | 3<br>2½ | Storage tanks<br>Deck-hose connections | 2½<br>2½<br>2½ | Service-pump supply<br>Storage tanks.<br>Deck-hose connections. | One each fireroom. |

TABLE III.—DATA FROM STANDARDIZATION TRIAL OFF ROCKLAND, ME., JULY 5, 1913.

| ..... | | | | | Starboard turbine. | | | | | Port turbine. | | | | | R.P.M. | | | S.H.P. | | | Averages by groups. | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | | Mer. gages. | | | | | Mer. gages. | | | | | | | | | | |
| Group No. | Run No. | Direction. | Speed, knots. | Nozzles open. | Chest pressure, pounds abs. | First stage, lbs. absolute. | Fourth stage, lbs. absolute. | Turbine exhaust. | Condenser. | Chest pressure, pounds abs. | First stage, lbs. absolute. | Fourth stage, lbs. absolute. | Turbine exhaust. | Condenser. | Starboard. | Port. | Mean. | Starboard. | Port. | Total. | Knots. | R.P.M. | S.H.P. |
| ... | 1 | S. | 11.696 | 2 | 195 | 12.5 | 5 | 27.2 | 27.95 | 185 | 13 | 5 | 27.2 | 27.95 | 191.39 | 193.20 | 192.30 | 328 | 240 | 568 | ... | ... | ... |
| 1 | 2 | N. | 12.461 | 2 | 190 | 12.5 | 4 | 27.2 | 27.95 | 175 | 12.5 | 4 | 27.2 | 27.95 | 196.87 | 197.57 | 197.22 | 361 | 269 | 630 | 12.115 | 196.01 | 612 |
| ... | 3 | S. | 11.842 | 2 | 190 | 12.5 | 4 | 27.2 | 27.95 | 175 | 12.5 | 4 | 27.2 | 27.95 | 197.03 | 197.66 | 197.35 | 355 | 264 | 619 | ... | ... | ... |
| ... | 4 | N. | 16.043 | 3 | 195 | 18 | 6.5 | 27.95 | 28.0 | 185 | 18 | 6.5 | 27.9 | 27.9 | 252.06 | 252.59 | 232.33 | 715 | 613 | 1,328 | ... | ... | ... |
| 2 | 5 | S. | 15.000 | 3 | 195 | 18 | 6.5 | 27.95 | 28.0 | 185 | 18 | 6.5 | 27.9 | 27.9 | 251.91 | 252.13 | 252.02 | 700 | 598 | 1,298 | 15.534 | 251.90 | 1,304 |
| ... | 6 | N. | 16.093 | 3 | 195 | 18 | 6.5 | 27.95 | 28.0 | 180 | 18 | 6'5 | 27.9 | 27.9 | 250.62 | 251.80 | 251.21 | 696 | 597 | 1,293 | ... | ... | ... |
| ... | 7 | S. | 17.266 | 4 | 175 | 22 | 8 | 28.2 | 28.2 | 165 | 22 | 8 | 27.95 | 27.95 | 291.67 | 294.41 | 293.04 | 1,103 | 995 | 2,098 | ... | ... | ... |
| 3 | 8 | N. | 18.237 | 4 | 175 | 21 | 8.1 | 28.2 | 28.2 | 165 | 21 | 8.1 | 27.95 | 27.95 | 290.03 | 290.76 | 290.40 | 1,082 | 862 | 1,944 | 17.81 | 292.62 | 2,020 |
| ... | 9 | S. | 17.501 | 4 | 180 | 22 | 8.5 | 28.2 | 28.2 | 165 | 22 | 8.5 | 27.95 | 27.95 | 295.81 | 297.42 | 296.62 | 1,136 | 959 | 2,095 | ... | ... | ... |
| ... | 10 | N. | 21.978 | 5 | 215 | 23 | 12.5 | 27.8 | 28.2 | 205 | 23 | 12.5 | 27.7 | 27.8 | 353.30 | 357.40 | 355.35 | 1,920 | 1,864 | 3,784 | ... | ... | ... |
| 4 | 11 | S. | 20.857 | 5 | 210 | 30 | 12.5 | 27.8 | 28.2 | 195 | 32 | 12.5 | 27.7 | 27.8 | 352.37 | 355.68 | 354.03 | 1,957 | 1,813 | 3,770 | 21.401 | 354.66 | 3,782 |
| ... | 12 | N. | 21.911 | 5 | 215 | 32 | 12.5 | 27.8 | 28.2 | 200 | 33 | 12.5 | 27.7 | 27.8 | 354.53 | 355.91 | 355.22 | 2,011 | 1,793 | 3,804 | ... | ... | ... |
| ... | 13 | S. | 23.033 | 6 | 245 | 45 | 17 | 27.6 | 28.1 | 230 | 42 | 17 | 27.8 | 27.8 | 396.65 | 399.95 | 398.30 | 2,906 | 2,773 | 5,679 | ... | ... | ... |
| 5 | 14 | N. | 23.936 | 6 | 255 | 45 | 16.5 | 27.6 | 28.1 | 235 | 43 | 16.5 | 27.8 | 27.8 | 394.14 | 401.81 | 400.48 | 2,934 | 2,810 | 5,744 | 23.488 | 399.80 | 5,490 |
| ... | 15 | S. | 23.047 | 6 | 245 | 45 | 16.5 | 27.6 | 28.1 | 235 | 43 | 16.5 | 27.8 | 27.8 | 399.01 | 400.43 | 399.72 | 2,994 | 2,800 | 5,794 | ... | ... | ... |
| ... | 16 | N. | 26.297 | 8 | 255 | 64 | 23 | 27.8 | 27.95 | 255 | 63 | 24. | 27.7 | 27.7 | 450.64 | 458.57 | 454.62 | 4,101 | 4,483 | 8,584 | ... | ... | ... |
| 6 | 17 | S. | 25.388 | 8 | 255 | 63 | 23 | 27.8 | 27.95 | 255 | 62 | 23.5 | 27.7 | 27.7 | 447.93 | 454.91 | 451.42 | 4,050 | 4,475 | 8,525 | 25.945 | 456.50 | 8,730 |
| ... | 18 | N. | 26.706 | 8 | 260 | 65 | 23 | 27.8 | 27.95 | 260 | 65 | 23.5 | 27.7 | 27.7 | 463.53 | 476.18 | 468.56 | 4,519 | 4,768 | 9,287 | ... | ... | ... |
| ... | 19 | S. | 27.907 | 12 | 250 | 95 | 35 | 27.25 | 27.4 | 240 | 90 | 33 | 27.3 | 27.5 | 540.37 | 539.21 | 539.79 | 6,674 | 6,167 | 12,841 | ... | ... | ... |
| 7 | 20 | N. | 28.571 | 12 | 253 | 90 | 36 | 27.25 | 27.4 | 235 | 93 | 32 | 27.3 | 27.5 | 536.79 | 538.34 | 537.82 | 6,645 | 6,236 | 12,881 | 28.255 | 538.80 | 12,848 |
| ... | 21 | S. | 27.972 | 12 | 253 | 95 | 36 | 27.25 | 27.4 | 240 | 88 | 32 | 27.3 | 27.5 | 540.54 | 538.96 | 539.75 | 6,611 | 6,180 | 12,791 | ... | ... | ... |
| ... | 24 | N. | 29.925 | 15 | 235 | 115 | 42 | 26.4 | 27.7 | 235 | 110 | 42 | 26.6 | 27.6 | 582.66 | 594.66 | 588.65 | 7,609 | 7,681 | 15,290 | ... | ... | ... |
| ... | 25 | S. | 29.557 | 15 | 240 | 110 | 41.5 | 26.4 | 27.6 | 240 | 110 | 43 | 26.45 | 27.6 | 590.35 | 601.06 | 595.71 | 7,779 | 7,950 | 15,729 | ... | ... | ... |
| 8 | 26 | N. | 29.90 | 15 | 240 | 110 | 41 | 26.4 | 27.6 | 240 | 110 | 41 | 26.6 | 27.6 | 586.69 | 601.59 | 594.14 | 7,696 | 7,630 | 15,326 | 29.818 | 596.50 | 15,606 |
| ... | 27 | S. | 29.801 | 15 | 245 | 113 | 42 | 26.3 | 27.5 | 245 | 113 | 44 | 26.45 | 27.6 | 596.65 | 606.01 | 601.33 | 7,897 | 7,990 | 15,887 | ... | ... | ... |
| ... | 28 | N. | 30.10 | 15 | 245 | 113 | 43 | 26.35 | 27.6 | 245 | 113 | 43 | 26.6 | 27.6 | 595.24 | 606.64 | 600.96 | 8,019 | 7,837 | 15,856 | ... | ... | ... |

Vacuum readings corrected to 30 inches barometer. Barometer readings, 29.8 inches.

## TABLE IV.—ENGINE-ROOM DATA.

| | July 5 | | July 7 | | July 8 | | July 7 | |
|---|---|---|---|---|---|---|---|---|
| Date of trial, 1913 | | | | | | | | |
| Duration of trials, hours | 4 | | 4 | | 4 | | 4 | |
| | Stb'd | Port | Stb'd | Port | Stb'd | Port | Stb'd | Port |
| Revolutions per minute | 567.17 | 575.48 | 407.62 | 414.74 | 249.61 | 251.79 | 192 | 197.1 |
| Average revolutions per minute | 571.33 | | 411.13 | | 250.70 | | 194.5 | |
| Speed, knots | 29.143 | | 24.036 | | 15.465 | | 12.033 | |
| Number of turbine nozzles open | 14 | 14 | 7 | 7 | none | | none | |
| Pressures: | | | | | | | | |
| Main steam, pounds gage | 236.8 | | 244.5 | | 230.6 | | 199.4 | |
| Cruising eng. H.P. cyl. valve chest, lbs. gage. | ... | ... | ... | ... | 203.8 | 206.9 | 116.1 | 116.2 |
| receiver, lbs. gage | ... | ... | ... | ... | 54.75 | 51.63 | 24.25 | 23.63 |
| exhaust, lbs. abs | ... | ... | ... | ... | 14.94 | 14.68 | 9.24 | 9.26 |
| Turbine steam chest, pounds absolute | 240.6 | 240.6 | 254.5 | 254.2 | ... | ... | ... | ... |
| first stage, pounds absolute | 104.3 | 103.2 | 50.0 | 51.6 | 12.16 | 12.04 | 9.55 | 9.55 |
| fourth stage, pounds absolute | 37.73 | 37.98 | 16.2 | 16.6 | 4.98 | 5.02 | 2.7 | 3.51 |
| eighteenth stage, pounds absolute | 22.14 | 22.45 | 13.24 | 12.95 | 2.91 | 3.35 | ... | ... |
| exhaust, inches of mercury* | 26.42 | 26.58 | 27.66 | 27.76 | 27.97 | 28.0 | 28.4 | 28.3 |
| Condenser vacuum top, inches of mercury* | 27.53 | 27.62 | 28.06 | 28.06 | 28.07 | 28.07 | 28.4 | 28.4 |
| Barometer, as read, inches of mercury | 29.92 | | 29.44 | | 29.73 | | 29.40 | |
| Auxiliary exhaust, pounds gage | 9.9 | | 3.4 | | 2.4 | | 1.7 | |
| Temperatures, degrees F: | | | | | | | | |
| Injection | 64.3 | | 59.2 | | 52.6 | | 55.1 | |
| Discharge | 91.2 | 90 | 85.7 | 87.1 | 68.9 | 64.7 | 62.5 | 64 |
| Wet-air pump discharge | 84.3 | | 76.4 | | 56.9 | | 59 | |
| Lubricating oil entering cooler | 135.3 | | 110.8 | | 91.6 | | unreliable | |
| leaving cooler | 126.7 | | 106 | | unreliable | | 95 | |
| Revolutions or double strokes per minute: | | | | | | | | |
| Main air pumps | 34.8 | 34.5 | 26.5 | 26.5 | 29.6 | ... | 30.7 | ... |
| circulating pumps | 219.6 | 221.5 | 133.4 | 131.8 | 113.1 | 115.5 | 119.5 | 114.4 |
| feed pumps | 28.9 | 28.9 | not used | | not used | | not used | |
| Forward fire and bilge pump | 38.3 | | 36 | | 37 | | 34 | |
| Forced-lubrication pump | 47.7 | | 38.5 | | 39.7 | | 29.5 | |
| Oil-cooler circulating pump | 8 | | 7.7 | | 6 | | 3.4 | |
| Miscellaneous: | | | | | | | | |
| I.H.P. of cruising engines | not used | | not used | | 407 | 439 | 237 | 214 |
| Total I.H. P. of cruising engines | ... | | ... | | 846 | | 451 | |
| Torsionmeter readings, inches | 2.13 | † | 1.35 | 1.367 | .40 | .48 | .33 | .31 |
| S.H.P. | 7,139 | 7,115 | 3,354 | 3,345 | 588 | 716 | 372 | 358 |
| Total S.H.P. | 14,254 | | 6,699 | | 1,304 | | 730 | |
| Displacement at middle of trial, tons | 1,057 | | 1,055.3 | | 1,060.5 | | 1,053.3 | |

*Corrected to 30-inch barometer.

†On the port torsionmeter one of the torsionmeter points broke off at the beginning of the trial.

The S.H.P. for port shaft was taken from the r.p.m. and S.H.P. curve obtained on the standardization trial.

## TABLE V.—FIREROOM DATA.

| | 29.143 | | | | 24.036 | | | | 15.465 | 12.033 |
|---|---|---|---|---|---|---|---|---|---|---|
| Speed, knots | | | | | | | | | | |
| Fireroom | Foward | | After | | Foward | | After | | After | After |
| Boiler | A | B | C | D | A | B | C | D | C | D |
| pressure, pounds gage | 255.8 | 258.3 | 255.2 | 257.6 | 252.2 | 257.4 | 256.1 | 257 | 242.6 | 201.2 |
| Number of burners used | 9 | 9 | 9 | 9 | 5 | 5 | 5 | 5 | 4 | 3 |
| Fuel-oil pressure at burners, lbs. gage | 201 | 177 | 159.4 | 148.5 | 234 | 197 | 182 | 174 | 154 | 158.5 |
| temperature at burners, °Fahr. | 76.4 | 76.4 | 74.8 | 81.1 | 74 | 74 | 71 | 76 | 73.9 | 83 |
| R.P.M. of blowers starboard | 1,367 | | 1,310 | | 822.5 | | not used | | 790 | 755 |
| port | 1,336 | | 1,283 | | 970 | | 1,205 | | not used | not used |
| Air pressure, inches of water | 4.76 | | 4.11 | | 2.8 | | 2.4 | | 1.7 | 1.6 |
| Fireroom temperature, degrees F | 103.4 | | 103.8 | | 85 | | 106.4 | | 95 | 99 |
| Feed-heater shell pressure, lbs. gage | 6 | | 6.1 | | not used | | not used | | not used | not used |
| temperature, degrees F | 150.6 | | 132.6 | | 71.1 | | 73.2 | | 62.2 | 63 |
| Fuel-oil pressure pumps, D.S.P.M. | 8.5 | 13.6 | 11.2 | 10 | ... | 11.6 | ... | 11.9 | 4.2 | 4.4 |
| booster pumps, D.S.P.M. | 70.4 | | not used | | 59.4 | | not used | | | |
| Auxiliary feed pumps, D.S.P.M. | not used | | | | 14 | | 14.2 | | 8.6 | 7.6 |
| Stack temperatures, degrees F | 488 | 507 | 567 | 582 | 431 | 412 | 422 | 443 | 400 | 372 |

*As needed for filling oil-measuring tanks.

TABLE VI.—ECONOMIC RESULTS.

| | | | | |
|---|---|---|---|---|
| Speed, knots | 29.143 | 24.036 | 15.465 | 12.033 |
| Auxiliary exhaust led to | 18th stage* | 18th stage | 1st stage | 1st stage |
| Average fuel-oil consumption, lbs. per hour | 18,528 | 9,241 | 2,032 | 1,282 |
| Average fuel-oil consumption, lbs. per knot at maintained speed | 635.78 | 384.47 | 131.41 | 106.57 |
| Average fuel-oil consumption, lbs. per knot at contract speed | 629 | 384.4 | 137.5 | 106.5 |
| Fuel-oil consumption guaranteed in contract, lbs. per knot | 686 | 402 | 141 | 110 |
| Fuel-oil consumption less than guar., lbs. per knot | 57 | 17.6 | 9.5 | 3.5 |
| in per cent. of guaranteed | 8.3 | 4.38 | 6.74 | 3.18 |
| Total steam consumption, pounds per hour | 242,554 | 115,833 | 26,405 | 15,134 |
| pounds per S.H.P. of main engines | 17.02 | 17.29 | 20.25 | 20.73 |
| reserve feed water used on trial, pounds | 8,476 | 8,284 | 1,792 | 1,940 |
| Pounds of oil burned per square foot of heating surface | .862 | .43 | .378 | .2385 |
| Pounds of water actually evaporated per pound of oil, including make-up feed | 13.20 | 12.76 | 13.21 | 12.18 |

* And feed heaters.

# THE EFFECT OF MIXING OILS IN FORCED-LUBRICATION SYSTEMS.

By Lieutenant G. S. Bryan, U. S. Navy.

Trouble has been experienced in the forced-lubrication systems of main engines and turbines in the Service from time to time, and it has frequently been suggested that this trouble might possibly be due to the mixing of two or more oils in the same system. In order to decide whether or not mixing oils had any harmful effect, the Bureau of Steam Engineering directed that a series of tests be undertaken at the Engineering Experiment Station at Annapolis with that end in view. The oils selected for the test were those that were submitted for the annual contract for 1913. In all there were eight oils, and of these five were paraffine base and three asphalt-base oils.

These oils were arranged in a list so that no two oils from the same company would be adjoining each other. Each one was then tested alone and in combination with the oils next adjoining it on the list. Thus:

Oil A alone.

Oil A in combination with oil B.

Oil B.

Oil B in combination with oil C.

Finally a mixture of all eight of the oils was tried.

The test was divided into three general parts:

(*a*) Test in oil-testing machine.

(*b*) Chemical examination for condition of refinement and determination of the physical constants before and after use in above machine.

(*c*) Test of oils mixed with steam and water.

A study of the oil-testing machines on the market failed to discover any one that was considered suitable for reproducing forced-lubrication conditions, and it was necessary to construct one at the Experiment Station.

The machine constructed has a steel journal, six inches long and one foot in circumference, turning in a regular white-

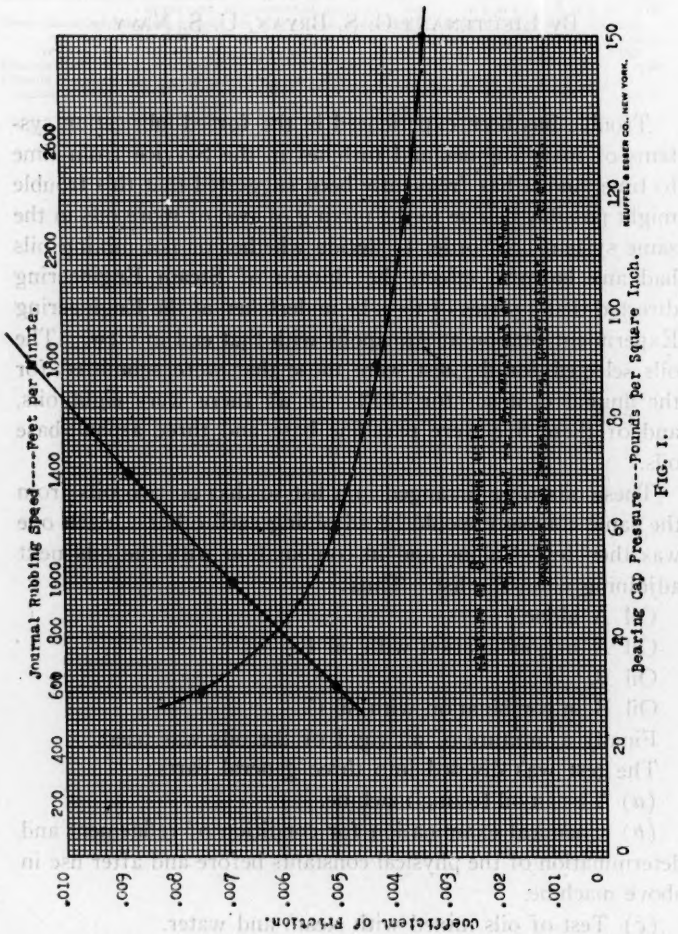



FIG. 1.

metal bearing, and lubricated by forced feed. It is constructed so that it is possible to regulate the rubbing speed of the journal, the temperatures of the bearing and of the entering oil, the pressure on the bearing cap, and the pressure on the forced-lubrication system. The journal is driven by an electric motor, and the power used up in friction is measured

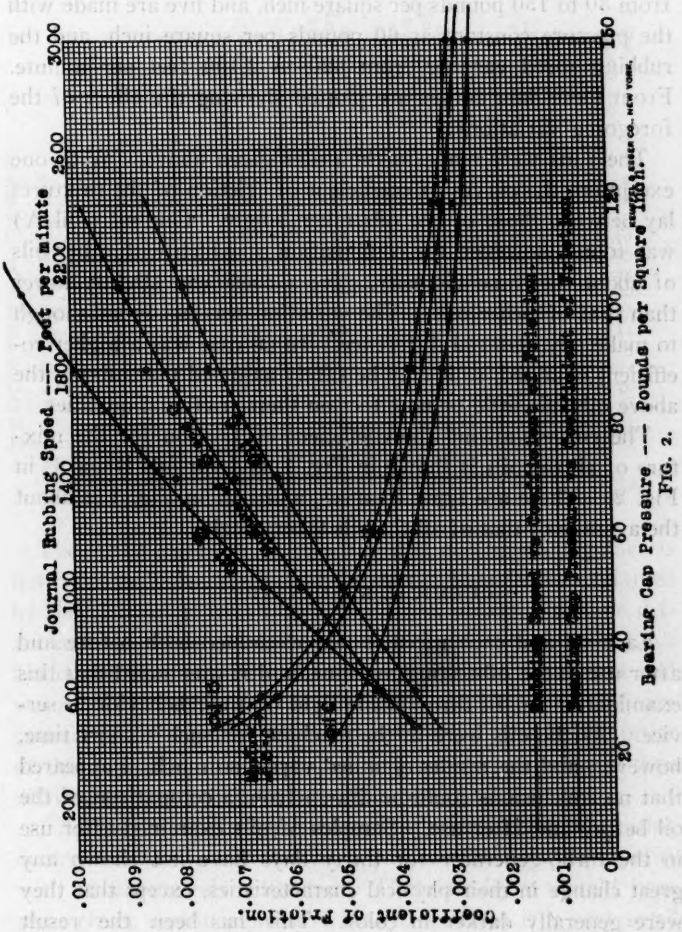



FIG. 2.

electrically, thus giving the data necessary to figure the coefficient of friction.

In running tests it is usual to keep the bearing temperature constant at 135 degrees F., which is assumed as an average working heat. Five 100-minute runs are made with each oil tested, at a speed of 620 r.p.m., with the pressure varying from 30 to 150 pounds per square inch, and five are made with the pressure constant at 60 pounds per square inch, and the rubbing speed varying from 620 to 2,100 feet per minute. From these data curves are plotted showing the effect of the foregoing variations.

The runs were made in the order given above. With one exception, the curves of coefficient of friction of the mixtures lay between those of the oils taken singly. One oil (Oil A) was found, however, the combination of which with other oils of about the same viscosity, gave a coefficient slightly lower than that of either one. The variation was not great enough to make any material difference. In no case was a higher coefficient obtained, so it may be safely assumed that mixing the above oils has little or no effect on their frictional qualities.

The curve showing the coefficient of friction for the mixture of all the oils is shown in Fig. 1, and for oils B and C in Fig. 2. These are typical curves, and that in Fig. 1 is about the average of that of all the oils taken singly.

## CHEMICAL EXAMINATION.

Each oil and each mixture was examined both before and after use in the oil-testing machine, and it was hoped that this examination might throw some light on their behavior in service. As the oils were in the machine for such a short time, however, and the results obtained varied so much, it appeared that no conclusions could be drawn from a comparison of the oil before and after use. Samples of oils examined after use in the turbo-generator for thirty days have not shown any great change in their physical characteristics, except that they were generally darker in color. This has been the result

found at other places, and it appears that it will take a long time for an oil to " wear out" in service.

The specific gravity, viscosity, flash and fire points and cold point are shown in Table I.

TABLE I.—PHYSICAL CONSTANTS OF OILS AND MIXTURES OF OILS.

| Oil. | Specific gravity. | Flash point, closed cup, degrees F. | Fire point, closed cup, degrees F. | Viscosity, 150 degrees F. | Cold point, degrees F. |
|---|---|---|---|---|---|
| A | .935 | 305 | 356 | 2.46 | —5 |
| A+B | .920 | 318 | 370 | 1.90 | —4 |
| B | .863 | 353 | 412 | 3.10 | 13 |
| B+C | .867 | 387 | 443 | 2.84 | 13 |
| C | .864 | 378 | 417 | 2.29 | 18 |
| C+D | .904 | 367 | 428 | 2.63 | 23 |
| D | .930 | 327 | 428 | 4.23 | 23 |
| D+E | .922 | 354 | 407 | 4.11 | 20 |
| E | .898 | 386 | 444 | 4.06 | 23 |
| E+F | .879 | 392 | 435 | 2.34 | 14 |
| F | .889 | 396 | 443 | 2.86 | 5 |
| F+G | .896 | 380 | 432 | 3.79 | 22 |
| G | .926 | 386 | 431 | 4.89 | 20 |
| G+H | .911 | 373 | 431 | 3.42 | 28 |
| H | .910 | 407 | 462 | 2.61 | 25 |
| H+A | .915 | 343 | 401 | 2.57 | 12 |
| A+B+C+D+E+F +G+H | .898 | 369 | 426 | 2.55 | 19 |

The specific gravity of two oils that are mixed together is not necessarily the average of the two. This fact is explained by the manner in which the oils diffuse, giving a volume different from the sum of the volumes of the straight oils. An analogous example may be cited in a mixture of alcohol and water. If several oils of different specific gravities are poured gently together they settle out into layers in the order of their specific gravities, but if they are stirred well and then allowed to stand they do not settle out at all.

The specific gravity has no apparent effect on the lubricating quality of an oil, and also no apparent effect on the rapidity with which the oils settle out. Theoretically it should have some effect, and it is probable that it does, but it is very small

in comparison with the other factors that govern the separation. It is convenient as a means of identifying an oil.

In general, the flash and fire points of mixtures were about the average of those of the straight oils. A general average of the eight straight oils, the eight mixtures, and the mixture of all eight oils is as follows:

| | Flash point, closed cup. | Fire point, closed cup. |
|---|---|---|
| Average of eight straight oils..... | 361 | 421 |
| Average of eight mixtures (of two oils) | 367 | 421 |
| Mixture of all eight oils..... | 369 | 426 |

The cold point was changed but little, either by mixing the oils or by use in the machine. This quality would have little bearing on a turbine oil as long as it is within the limits found for all the oils tested.

The viscosity was determined with the Engler-Ubbelohde viscosimeter. It was found that the viscosity of a mixture generally lay between those of the straight oils, the exceptions being in the cases where the coefficient of friction was also less. The viscosity was determined at 90, 120, 150, 180, 212 and 300 degrees F. In the table only the viscosity at 150 degrees F. is shown.

All the oils were tested for acidity and for the presence of moisture, hard and soft asphalt, fatty oils, sulphur, charring constituents, emulsifying constituents, wax-like constituents, napthenic acids, sulphonated oils, soaps and resin. Most of the oils were free from all the foregoing constituents. One oil was found that emulsified very readily and another that had a trace of hard asphalt in it. Traces of these products were found in every mixture made with these oils. The oil that emulsified was oil F, and the one that contained the trace of asphalt was oil H.

### MIXTURES WITH STEAM AND WATER.

Each oil and each mixture was given the test outlined below. Fifty c.c. of the oil was mixed with each of the following: (*a*) 50 c.c. of distilled water at 65 degrees F.

(*b*) 50 c.c. of Severn River water (340 grs. chlorine) at 65 degrees F.

(*c*) 50 c.c. of boiling distilled water.

These mixtures were stirred in a pint commercial fruit jar for five minutes, and were then poured in graduated glass cylinders and allowed to settle out at room temperature (70 degrees to 80 degrees F.), observations being made periodically of the total amount of oil and water that had not separated from the mixture.

Fifty c.c. of oil and 50 c.c. of distilled water were stirred in a steel-capped cylinder under 50 pounds' steam pressure, and 100 c.c. of the oil was stirred in the same cylinder under 300 pounds, steam pressure, observations being taken as before.

Table II shows the result of these observations. The percentages shown represent the amount of "separated matter," which is the general term for everything except clear water and good oil which has separated out.

In most cases the combinations show percentages about the mean of those of the straight oils. In some cases the rate of separation appears to be a little slower, but different conditions of temperature under which the oils settled out would be enough to account for this slight variation and, from a practical point of view, no harmful results are indicated by this part of the test.

The exact nature and cause of this "separated matter" is not known. It occurs in various forms from a light, spongy, flocculent material to a thick, creamy emulsion. The latter form is probably mostly a simple emulsion. Most of it will generally settle out into good oil and clear water if it is allowed to stand indefinitely, but some of the separated matter never settles out. Whether it is waxy in character or is caused by sodium salts of napthenic acids, as some writers claim, there is no doubt that it should not be present. With the present knowledge at hand, about all that can be said is that some action, whether mechanical or chemical, occurs between the water and the oil when the mixture is thoroughly

TABLE II.—PERCENTAGES OF SEPARATED MATTER WITH MIXTURES WITH STEAM AND WATER.

| Oil. | Distilled water. | | | | | Salt water. | | | | | Boiling distilled water. | | | | | Low-pressure steam. | | | | | High-pressure steam. | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Percentage of separated matter at end of—hrs. | | | | | Percentage of separated matter at end of—hours. | | | | | Percentage of separated matter at end of—hours. | | | | | Percentage of separated matter at end of—hours. | | | | | Percentage of separated matter at end of—hours. | | | | |
| | ½ | 1 | 3 | 5 | 24 | ½ | 1 | 3 | 5 | 24 | ½ | 1 | 3 | 5 | 24 | 24 | 48 | 72 | 96 | 120 | 24 | 48 | 72 | 96 | 120 |
| A | ½ | ½ | ½ | ½ | ½ | 52* | 50* | Trace. | ... | Trace. | ½ | ½ | ½ | ½ | ½ | 7 | 6 | 6 | 5 | 5 | ... | ... | ... | 15 | 15 |
| A + B | ½ | ½ | ½ | ½ | ½ | 52* | 50* | Trace. | ... | Trace. | 1 | ½ | ½ | ½ | ½ | 8 | 8 | 8 | 6 | 6 | ... | 10 | 10 | 10 | 10 |
| B | ½ | ½ | ½ | ½ | ½ | ½ | ½ | Trace. | ... | Trace. | 1 | 1 | 1 | 1 | 1 | 7 | 6 | 6 | 6 | 6 | ... | ... | ... | 15 | 15 |
| B + C | ½ | ½ | ½ | ½ | ½ | ½ | Trace. | Trace. | ... | Trace. | 1 | 1 | 1 | 1 | 1 | 11 | 10 | 6 | 6 | 6 | ... | 10 | 10 | 10 | 10 |
| C | 2 | 1 | 1 | 1 | 1 | 2 | ½ | ½ | ½ | ½ | 1 | 1 | 1 | 1 | 1 | 4 | 4 | 4 | 4 | 4 | ... | ... | ... | 14 | 14 |
| C + D | 3 | 2 | 2 | 1 | ½ | 23 | 10 | Trace. | ... | Trace. | 1 | 1 | 1 | 1 | 1 | 13 | 10 | 7 | 6 | 6 | 10 | 10 | 8 | 6 | 6 |
| D | 3 | 2 | 2 | 1 | ½ | 55* | 55* | 2 | Trace. | Trace. | 50* | Trace. | ... | Trace. | Trace. | ... | 7 | 5 | 5 | 5 | ... | 11 | 8 | 8 | 6 |
| D + E | 3 | 2 | 2 | 2 | 1½ | 62 | 60 | 3 | Trace. | Trace. | 50* | Trace. | ... | Trace. | Trace. | ... | 4 | 3 | 3 | 3 | ... | 6 | 5 | 5 | 4 |
| E | 67 | 15 | 7 | 2 | 1½ | 91 | 81 | 20 | Trace. | Trace. | 52* | 1 | 1 | 1 | 1 | 9 | 7 | 6 | 6 | 6 | ... | 6 | 6 | 5 | 4 |
| E + F | 56 | 10 | 8 | 8 | 7 | 98 | 90 | 60 | 30 | 2½ | 92 | 89 | 85 | 79 | 50 | ... | 11 | 9 | 7 | 6 | ... | ... | 30 | 15 | 8 |
| F | 62 | 57 | 15 | 14 | 12 | 71 | 69 | 32 | 12 | 10 | 66 | 62 | 60 | 57 | 54 | ... | 17 | 15 | 13 | 11 | ... | ... | 12 | 12 | 11 |
| F + G | 56 | 56 | 8 | 8 | 6 | 100 | 98 | 50 | 40 | 4 | 59 | 57 | 50 | 46 | 45 | ... | 14 | 12 | 7 | 6 | ... | ... | 14 | 11 | 11 |
| G | 57 | 55 | 53 | 1 | 1 | 100 | 96 | 25 | 10 | ½ | ½ | ½ | ½ | ½ | ½ | ... | 22 | 17 | 15 | 12 | ... | ... | ... | 20 | 12 |
| G + H | 59 | 9 | 8 | 8 | 1 | 97 | 75 | 5 | 1 | Trace. | 53 | ½ | Trace. | Trace. | Trace. | ... | 10 | 8 | 6 | 5 | ... | ... | 11 | 11 | 11 |
| H | 55 | 5 | 5 | 5 | 1½ | 83 | 70 | 25 | 3 | Trace. | 1 | Trace. | Trace. | Trace. | Trace. | 11 | 8 | 8 | 8 | 7 | ... | ... | ... | 25 | 17 |
| H + A | 64 | 21 | 16 | 3 | 1 | 63 | 63 | 12 | 3 | Trace. | 2 | 2 | ½ | ½ | Trace. | 9 | 9 | 8 | 8 | 8 | ... | ... | ... | 27 | 22 |
| A + B + C + D + E + F + G + H | 3 | 3 | 1½ | 1½ | 1 | 90 | 35 | 3 | 1 | ½ | ½ | Trace. | Trace. | Trace. | Trace. | 10 | 9 | 7 | 6 | 6 | ... | ... | 15 | 14 | 14 |

* Indicates that water had all settled out but the mixture was cloudy and held water in suspension.

agitated, and that the quantity of this resultant matter, whatever it may be, depends upon the temperature of the oil and water when they are mixed, the degree of stirring to which they are subjected, and the temperature of the mixture while settling out.

Some of the matter formed at low temperatures is broken up into water and oil as the temperature is raised, and new combinations are formed. One oil (oil F), which emulsified very readily, was stirred for five minutes with equal volumes of water at different temperatures and allowed to settle out for 24 hours. At the end of that time, the following percentages of separated matter were found:

| Mixed at Temp. F. | Percentage of separated matter. |
|---|---|
| 51 | 1 |
| 65 | 3 |
| 83 | 5 |
| 120 | 26 |

This separated matter took the form of a thick jelly-like emulsion. The same character of emulsion was formed by bubbling steam through the oil for a few minutes. When the mixture was stirred under fifty pounds' steam pressure, however, this emulsion disappeared, and a different kind of separated matter made its appearance.

The water that settles out from the mixtures with water and steam is always of a milky color with distilled water and a thicker creamy color with steam, being generally thicker with the more viscous oils. With salt water the water is fairly clear. By mixing salt water with the milky and creamy colored water referred to the latter may be cleared and slightly more separated matter precipitated.

Tests are now being conducted in the friction machine to determine the change in friction due to adding fresh and salt water to both straight oils and mixtures of oils.

## CONCLUSIONS.

It is believed that the foregoing test has demonstrated that no harmful results are obtained due to mixing any or all of the oils tested. It is also believed that the eight oils tested are representative enough to cover the field of oils for forced lubrication, and that it may be safely assumed that the conclusions drawn from the test of the above oils will apply to all straight mineral oils for use in forced-lubrication systems.

# GEARING AND THE ELECTRIC DRIVE.

## PRESENT USE AND FUTURE DEVELOPMENT.

By Lieut. Commander H. C. Dinger, U. S. N., Member.

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Gearing is being applied in a great many places in vessels building for the United States Navy. The first vessel to use gearing in the United States was the Collier *Neptune*. The *Neptune* is being fitted with a turbine and condensing apparatus of new design which will be tried in the course of a few months. The gearing installed on the *Neptune* operated satisfactorily, but the original turbine installed was uneconomical, the combined result being that the turbine and reduction gear was not as economical as the commercial type of reciprocating engine installed on the sister ship, the *Cyclops*. With the new turbine, which is much more economical, a great improvement in economy is to be expected.

Gearing is being used for the cruising turbines on Destroyers Nos. 55, 57, 58, 59, 61 and 62, and gear on the main turbines is used on Destroyer 60 and on the submarine tenders *Bushnell* and *Melville*. Geared cruising turbines are also being employed on the battleship *Pennsylvania* and on No. 39.

Gearing has been very successful on fast passenger steamers built in England. So far as economy is concerned the use of geared turbines results in an appreciable gain over the reciprocating engine and the directly-connected turbine. Gearing has, however, not shown any superiority on the score of economy over systems employing the combination of reciprocating engines and L.P. turbines, the latest vessel employing this system being the *Britannic*. It is, of course, possible to employ an L.P. geared turbine in combination with a reciprocating engine; but this has as yet not been contemplated in any actual installation.

The chief advantage of gearing, electric drive or the hydraulic transmitter is the reduction of weight and space that can be made and the great reduction in the size of rotor. The use of a high-speed turbine also permits of superheat being employed and permits the possibility of reducing the carry-over losses to a considerable extent.

Gearing is being introduced in the British Navy for cruising turbines of the light cruisers of the 1912–1913 class. No exact information as to any actual installation of gearing for main turbines of British naval vessels is available, though the matter is under consideration, and no doubt some installations with gear drive for main turbines are actually being made.

The following Westinghouse gears are being manufactured.

The gear for the *Melville* will have two pinions on the horizontal center line, diametrically opposite each other, the pinion speed being 1,400 r.p.m. and the propeller revolutions 110. The total horsepower transmitted is nominally 2,000 per pinion. The *Pennsylvania* and Battleship No. 39 will each have two gears, which are connected to the main turbine shafts by disconnecting clutches, the cruising turbines being out of service at full power, in which case the cruising turbines and gear are not in use. Each of the *Pennsylvania's* gears will nominally transmit 1,400 S.H.P., and each has two pinions on the horizontal center line, the reduction being from 1,785 to 119 r.p.m. The gears for Battleship No. 39 are also for use at cruising speed only, the speed of the cruising turbine being 2,500 r.p.m. and that of the propeller 192 r.p.m. at cruising speed. The horsepower of each of the cruising turbines of Battleship No. 39 will be 2,000, and in this case a different arrangement of cruising turbines is used, there being only one cruising turbine on each shaft, instead of two, as in the *Pennsylvania,* and for each gear Battleship No. 39, therefore, has only one pinion. The Westinghouse Machine Company has at present orders for 50,000 H.P. of geared installations, while over 50,000 H.P. are in operation.

Gearing for turbine reduction work has been made in this

country by the DeLaval Turbine Company for the last twenty-five years. This company is manufacturing gears for the cruising turbines of some of the United States naval vessels.

Reduction gears are also being manufactured in the United States by the Falk Company, of Milwaukee, Wis., who are making the gears for some of the cruising turbines to be used on some of the new destroyers.

It will thus be seen that a number of firms in this country have the equipment for turning out gears to be employed in marine propulsion.

The electric drive as exemplified in the installation on the *Jupiter* and the Föttinger transmitter installed on the German passenger ship, *Königin Luise,* show a very high economy, but the result is due to the use of more economical turbines and boilers than have been installed on the geared turbine vessels. The geared turbine vessels have used low steam pressures and no superheat. On this account the channel steamers using Parsons turbines have not shown the full fuel economy that is possible with gearing.

The electric drive and Föttinger transmitter have an advantage in that no backing turbine is required, and also any power up to maximum is available for going astern. Furthermore, the turbine always runs in the same direction, which would offer less likelihood of blading troubles.

The relative advantages and disadvantages of these different systems are concisely stated in the article "Transmission of Propulsive Power in Ships," by Ernest N. Janson, appearing in the February, 1914, number of the Journal of this Society.

The rival installations being developed in America are represented by the experimental machinery installed on the naval colliers *Jupiter* and *Neptune.* These two installations have opened up new and novel developments for marine propulsion, and it is not unlikely that either or both of these systems may soon be employed on capital vessels.

The recent performance of the *Jupiter* is described in this number.

Points of advantage that these systems possess over the direct turbine drive. The following may be enumerated:

*Reduction in weight.*—This will probably amount to about 10–15 per cent. of the machinery installation. This percentage will be larger on very fast, high-powered vessels than on those of moderate power.

*Increase in economy,* especially at cruising speeds. This also will amount to about 10–15 per cent. An exact amount can not be stated, but the indications are that a substantial gain in economy will be obtained.

*Reduction in size of units.*—This is a very material military advantage. It will enable more spares to be carried, presents the possibility of quick repair in case of accident. A spare turbine should be built and held in readiness for installation in case of any accident or in case any of the turbines required extensive overhauling.

One of the serious disadvantages of direct turbine drive in large vessels is the enormous size of the turbine casings and the heavy and complicated tackle that is necessary to lift casings for internal examination. In order to make examinations the vessel is practically laid up for a month or more, and requires the assistance of dock-yard facilities.

*Saving of space.*—This is very material and is one of the great advantages of the indirect-drive system. This saving of space and the smaller units enable all parts to be placed in a more accessible arrangement which permits of better upkeep and maintenance.

The success of the systems will depend to a very great extent on the design of turbine. The following are some of the pertinent matters that should be embodied.

*Impulse blading for initial stages.*—The use of impulse blading is increasing. The Westinghouse system and the General Electric electric drive and the A. E. G. turbine used with the Föttinger transmitter all employ impulse blading. Practically all the late British construction employs impulse blading with both Parsons and Brown-Curtis turbines. The use of impulse

blading permits of using a much shorter and lighter turbine, enables superheat and high-pressure steam to be employed and, when the blading is properly designed for strength, practically eliminates blading troubles.

If 100 degrees superheat and steam of 200 pounds pressure are used, on the basis of the *Jupiter's* performance, a water consumption of 10 pounds per S.H.P. can be obtained, which will, with gearing, give about 1.1 pounds of coal per S.H.P., and with the electric drive about 1.2. Coal at 14,500 B.t.u. The *Jupiter's* turbine is about the most economical marine turbine yet tried, but it does not approach very closely to the best results obtained with similar turbines on shore. The Parsons marine geared turbines have as yet not done better than 12 pounds of water per S.H.P. This indicates that a well-designed impulse turbine can equal or even secure an advantage in economy over the Parsons.

A recent test of a large G. E. Curtis turbine generator with 200 pounds pressure, 120 degrees F. superheat and 29-inch vacuum gave 11.4 pounds of water per kw. This generator applied to the electric reduction gear would furnish a S.H.P. on 9 pounds of water, requiring about .92 pound of coal with marine boilers that might be used with such an installation.

Another advantage of the impulse type is that the turbine is not divided into high and low pressure.

The principal advantage of the Parsons lies in the fact that it has been fully developed in its mechanical details. This can not be said for the impulse turbines experimented with in this country. The troubles encountered with the Curtis and Zoelly types for marine work have been due chiefly to defective mechanical details. These have been largely overcome, but whether they have entirely has not been fully demonstrated. On shore installations the General Electric Curtis turbines have been fully developed, even perhaps to a greater degree of simplicity and reliability than has the Parsons turbine.

*Employ superheated steam.*—By using superheated steam 100 degrees F. an increase in economy of 10 per cent. can be secured. This should be employed on all classes of vessels;

the saving of fuel is important in all cases, and superheaters do not increase the weight of the installation.

*Employ efficient and simple condensing apparatus.*—Too much complication in this important matter has caused considerable trouble.

---

A comparison of gearing and electric drive on various points is made as follows. The advantage is not all in one way and the special conditions of vessel and her service requirements will govern the choice of one type or the other.

| *Gearing.* | *Electric Drive.* |
|---|---|
| **WEIGHT.** | |
| The advantage rests with the gearing, since the gear will not weigh as much as the generator and motor combined. | This saving of weight will, however, not be a very large percentage of the total machinery weight, probably not more than 1 or 2 per cent. |
| **SPACE.** | |
| The advantage will rest to a slight extent with gearing. | The space can, however, be in many cases more advantageously utilized. The generator can be placed at some distance from the motor. The shaft can be greatly shortened. |
| **ECONOMY.** | |
| With the same boilers and turbines gearing has an advantage of about 6–8 per cent. at full power. | Owing to ability to use only one generator at below half power, the ability to run generator at practically constant speed and the flexibility of the electric bond to adjust itself to any speed ratio, it is likely that the initial advantage of gearing in economy will disappear at cruising speeds. |
| **REVERSIBILITY BACKING POWER.** | |
| Requires a backing turbine which complicates design and enables only a portion of full power to be used for backing. Limited to about 50 per cent. backing power. | Full backing power is secured without disturbing turbine. Reversing is made particularly easy. |

## RACING.

| *Gearing.* | *Electric Drive.* |
|---|---|
| The gear and turbine will be subject to the racing action of the propeller. | There will be no racing. The motor will govern the propeller and maintain even speed. This fact may possibly produce a slight gain in propulsive efficiency. |

## HANDLING.

The advantage will probably rest with the electric drive.

## ADAPTABILITY TO BEST CONDITIONS FOR EFFICIENT PROPULSION.

The gear ratio having been adopted it can not be changed. If the effective H.P. and propeller efficiency have not been accurately estimated an improper ratio will be present which may prevent the turbine from securing its most efficient speed. Unless very high gear ratios are employed the most economical turbines can not be used.

The relative speed of propeller and turbine can be altered to any extent. The turbine can therefore run at its most efficient speed without reference to the propeller speed. This enables a very high-speed turbine to be used, which allows of greater economy.

The design can utilize the very best propeller conditions without affecting the design or limitations of the turbine.

## FACILITY FOR REPAIR.

No special advantage. Whole engine plant has to be run when underway.

In case of accident generator and motor parts could be renewed more easily than gear could be repaired. One generator can be repaired while other works to full power. This will give about ¾ speed.

## REPAIRS TO BE EXPECTED.

Repairs to gearing will be little or nothing; but if damaged to any extent would have to be replaced.

A spare gear should be at hand to replace one damaged by accident.

The repairs to generator and motor will probably be of considerably greater extent and expense than that to gear, but these repairs can be more readily made.

## CARE IN OPERATION.

| *Gearing.* | *Electric Drive.* |
|---|---|
| Little or no special care required for the gear. Oil supply the essential matter. | Considerable attention will have to be given to maintaining generator and motor in condition. This also requires a special knowledge of electrical features for which personnel will have to be specially trained. Ventilation of motor and keeping it dry will require special attention. |

## FACILITY FOR INSTALLATION.

| | |
|---|---|
| Location of turbines and gear fixed by position of shaft. | Generators can be located anywhere without regard to shaft. Best location of condenser, steam piping, etc., can be secured. This enables a more adaptable and simple arrangement to be made. |

## CERTAINTY OF DATA ON WHICH TO BASE DESIGN.

| | |
|---|---|
| Gears suitable for a very large power have not yet been built; each installation practically necessitates a new design of the gear. Existing installations have been successful. | All portions of the electric-drive apparatus have been built and fully tested out. They have been applied to one marine installation and have been successful. |

## SHAFT BREAKAGE.

This is not an infrequent happening in naval vessels, and the U. S. Navy has in the past few years had a considerable number of shaft failures. The reasons for shaft breakage have not been conclusively determined, and various authorities have attributed different causes for the trouble. Among these the following may be mentioned.

(1) Improper working or heating of the metal.

(2) Too small a factor of safety allowed, owing to lack of homogeneity of the metal.

(3) Unexpected overloads due to application of stresses greater than allowed for in the design.

(4) Improper balancing or support, by reason of which detrimental vibrations are set up which weaken the metal and finally cause fracture.

(5) Operation of engines at or near the critical vibration speed, which produces unusual stresses and causes crystallization and fatigue of metal.

Most of these troubles are eliminated by the use of turbines, but still when backing turbines are employed there is likely to be a sudden reversal of power which may have the effect of a sudden blow and cause an unusual stress. Similarly with gearing there is still the possibility of setting up vibrations. With the electric drive, however, there are no vibrations in the shaft and no periodic changes of speed which might cause vibrations. Nothing in the nature of a sudden blow or shock is brought upon the shafting.

## CHARACTER OF MACHINERY TO EMPLOY ON DIFFERENT TYPES OF VESSELS.

The particular advantage of the electric drive will be more evident on a large vessel, as on a battleship, than on a smaller one, as a destroyer. Also, since weight and space are of paramount importance on a destroyer as is also economy at high speeds, it appears that gearing should be used on destroyers: turbines to be of impulse type employing superheated steam. The reduction in weight and superior economy will permit the possibility of securing the same power and radius of action at high speeds with a decrease of about 30 per cent. of the total weight of machinery and fuel, as compared with the direct turbine drive. No great gain over results obtained at low speeds where cruising engines are employed is to be expected.

On a large vessel the better economy at cruising speeds, superior handling facilities and the possibility of being underway and only using one of the main generators, and the ability

to easily sustain a slight increase in weight and space, would make the electric drive a very attractive proposition.

Should it be adopted, the opportunity of standardizing the machinery of a group of four or six battleships should not be lost sight of. If it is adopted it should be used on at least six battleship of the same type.

It would appear that the above division of application would utilize both gearing and the electric drive, each in its most suitable field.

### FUTURE DEVELOPMENT—ELECTRIC DRIVE.

The electric drive as a general proposition also opens up a field for further improvement, especially for large naval vessels.

The following is a suggested application of the system in a manner that is quite promising. The parts suggested for use have to some extent all been actually tried, built and successfully operated. It is proposed to use the Diesel-engine generator in combination with steam-driven generator and electric drive. This installation when thoroughly and practically developed will enable special advantages to be secured for a large naval vessel.

Economy at a moderate speed is specially desired, while, on the other hand, there must be entire reliability, and the maneuvering qualities must be the very best.

There is as yet no prospect of the direct Diesel drive being used for very large powers, such as would be required on a battleship. The engines are as yet not sufficiently reliable and the maneuvering abilities are as yet unsatisfactory. The weight of the Diesel drive is also a serious objection.

Since the power at half speed and below is but a small portion of the total power, an arrangement whereby Diesel engines might be used for the slower speeds up to ¼ or ⅓ power, while the larger part of the power is produced by high-speed steam turbines, would enable the installation to be made without any excessive weight and would make it fully reliable and

the operation of the motors not necessarily dependent on the Diesel unit. The electric drive permits of this being made entirely possible.

With a total power of 36,000 S.H.P. install the following:

Two shafts with electric motors.

Two high-speed steam-driven generators of 12,000 horsepower each.

One Diesel-driven generator of 12,000 horsepower.

Boilers with superheaters for the steam-driven plant and for such auxiliaries as may be driven by steam.

The Diesel engine might be installed in two units connected to the same generator and employing 12 cylinders would bring the power per cylinder to 1,000, which is well within the realms of practicability. This Diesel engine could be built like a powerhouse Diesel engine, many of which are now in successful operation. It would therefore present no unusual complications and is an entirely feasible mechanical job. The steam plant and its two electric generators should be of the latest type developed. The use of the heat in the exhaust of the Diesel and of the piston circulation for feed heating and thus assist in generating steam should be worked out. This, if efficiently done, could probably improve the fuel consumption of the steam plant at least 10 per cent.

At low speeds, for economy, the Diesel unit would be operated. This would enable a speed of 15 to 16 knots to be obtained on a battleship with a consumption of about .4 pound of oil per S.H.P., or at about one-half to one-third the consumption with the ordinary steam plant. At higher speeds one steam generator and then the other would be brought into service.

At full speed the fuel consumption of the combined plant would be about .8 pound of oil per S.H.P. without considering the economy due to possible use of waste heat of the Diesel engine. It is very likely that the fuel consumption might be reduced to below .7 pound per S.H.P. by using the heat in the Diesel exhaust.

This system is suggested as a possibility. The principal

difficulty in the way of its adoption is that a reliable Diesel engine of the size required, capable of operating satisfactorily on cheap asphalt-base oil, has not yet been developed. If carefully worked out and all parts are developed, in the light of present information, it would secure a remarkable advantage over any results that are now obtained or likely to be secured in the near future.

It would enable the following to be done in comparison with the latest direct turbine drive:

1. Double the steaming radius of vessel at speeds below 16 knots.

2. Increase steaming radius above 16 knots about 25 per cent.

3. Enable two-thirds of the generating units to be overhauled and repaired while still permitting vessel to steam at over half speed. Either the Diesel engine or the steam-generating plant could be disconnected and still operate the vessel with full maneuvering power.

4. It would enable the engineering personnel to be reduced about 25 per cent.

5. The cost would probably be about the same as for a direct turbine drive of the same power.

6. Cost of fuel for operating vessel would be reduced to nearly one-half, provided that the Diesel engine will work on the cheap oils available.

# TORPEDO-BOAT DESTROYERS *BISSON* AND *RENAUDIN.*

By Ernest N. Janson, Member.

(Data from "Journal de la Marine Le Yacht.")

These two new destroyers of the French Navy, for the Mediterranean service, were built, so far as the construction of the hulls is concerned, at the government yards in Toulon; boilers and auxiliaries at l'etablissement d'Indret, and the turbines by La Maison Bréguet, of Paris.

The manner in which the figures below demonstrate the successful results obtained goes to prove the skill with which the designers and builders of these boats solved the problems laid down to them; this being especially true with regard to the turbines.

From the official trials the following results for the full-power runs were obtained:

| | *Bisson.* | *Renaudin.* |
|---|---|---|
| Displacement on trial, tons | 716 | 721 |
| Mean draught, feet | 8.41 | 8.46 |
| Draught aft, feet | 9.60 | 9.64 |
| Number of boilers used | 4 | 4 |
| Mean pressure at boilers, pounds per sq. in | 220.45 | 236.14 |
| turbine, pounds per sq. in | 170.67 | 179.2 |
| Air pressure in boiler rooms, ins. of water | 6.04 | 6.29 |
| Pressure at oil burners, lbs. per sq. in | 99.5 | 79.64 |
| Revolutions per minute, turbines | 612 | 599 |
| Mean speed of ship during 6 hours, knots per hr. | 31.05 | 30.55 |
| S.H.P. | 15,000 | 15,100 |
| Mean vacuum, main condensers, ins | 26.57 | 26.29 |
| Actual coal consumption, average hourly, total lbs. | 22,546 | 22,461 |
| Actual coal consumption, average hourly, per S.H.P. pounds | 1.503 | 1.487 |
| Coal consumption, average hourly allowed, total pounds | 26,490 | 26,490 |
| Condition of the sea | Light swell. | Rough. |

*Official Coal-Consumption Trial of 8 Hours Duration.*

| | *Bisson.* | *Renaudin.* |
|---|---|---|
| Trial displacement, tons | 712 | 711 |
| Number of boilers in use | 2 | 2 |
| Mean pressure at boilers, lbs. per sq. in | 213 | 216.18 |
| Mean pressure at turbines, lbs. per sq.in | 182.8 | 191.32 |
| Air pressure in boiler room, ins. water | 1.57 | 1.33 |
| Pressure in oil burners, lbs. per sq. in | 99.54 | 71.10 |
| Revolutions per minute, turbines | 239.7 | 241.5 |
| Mean speed, knots per hour | 14.03 | 14.264 |
| S.H.P. | 780 | 810 |
| Mean vacuum, main condenser, ins | 29.2 | 29.01 |
| Actual coal consumption, average hourly, total pounds | 2,954 | 2,342 |
| Actual coal consumption, average hourly, per S.H.P., pounds | 3.78 | 2.89 |
| Condition of the sea | Calm. | Calm. |

These destroyers are quite similar with those built for other navies, and lately those for the French Navy, both as regards the construction of hull and the general arrangement, also armament; they differ only in the arrangement and type of turbines.

Principal dimensions are:

| | |
|---|---|
| Length over all, feet | 255.34 |
| Breadth, feet | 25.84 |
| Depth, feet | 16.73 |
| Draught aft, fully loaded, feet | 9.84 |
| Displacement on full load draught, tons | 800 |

The ship lines are very fine, a high freeboard forward being incorporated in order to maintain speed in a relatively high sea; the stem is made of forged steel, shaft struts of cast steel, the plating of Siemens steel.

The hull is divided longitudinally in ten water-tight compartments, the first three of which are occupied by collision compartments, the galley, ammunition, store rooms and crews' quarters; the fourth and fifth compartments are occupied by the boilers, the next two compartments being taken by the turbine installation; finally the eighth and ninth compartments contain quarters for officers and petty officers, the tenth being the forward collision compartment.

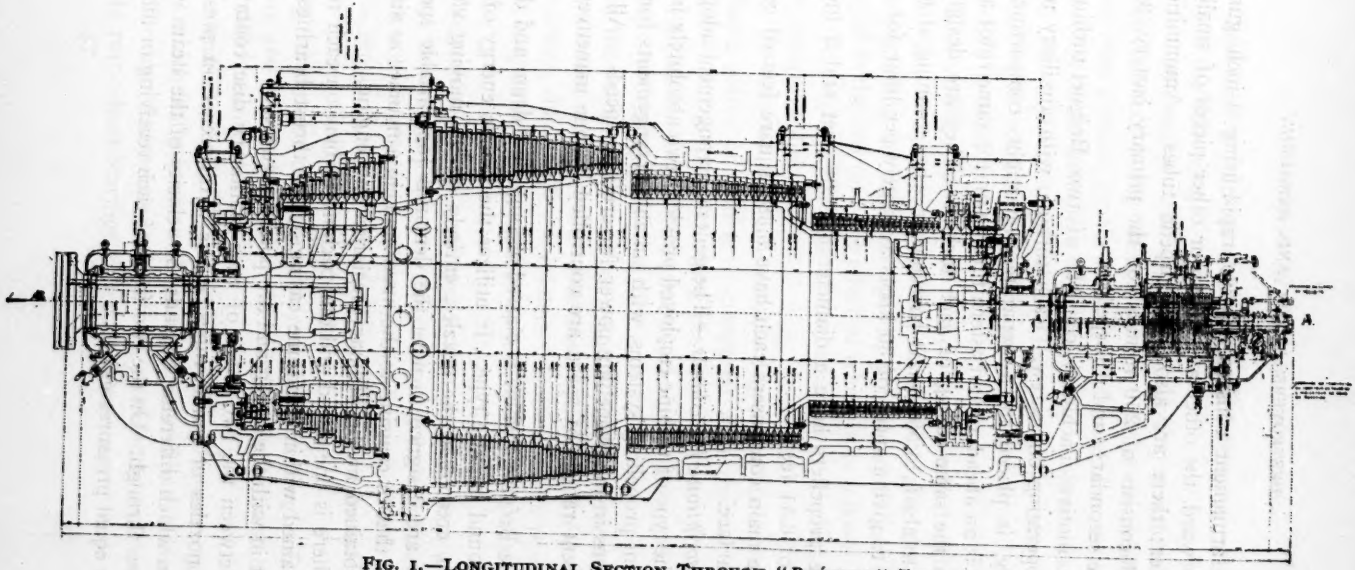


FIG. 1.—LONGITUDINAL SECTION THROUGH "BRÉGUET" TURBINE.

The armament consists of two rapid-firing 4-inch guns, one forward, the other aft, and four other pieces of smaller diameter; there are also four torpedo tubes. Ammunition carried consists of 250 rounds for the primary battery, 350 for the secondary, and 6 torpedoes.

The motive machinery consists of two Bréguet turbines, each operating one screw. Each turbine, with auxiliary machinery, is placed in a separate watertight compartment. There is an ahead and an astern turbine on the same rotor and within the same casing of each turbine. They are designed for a total of 14,500 shaft horsepower when running at 630 r.p.m. to give a speed of 30 knots averagely per hour for six hours.

The propellers have a diameter of 7.08 feet and a mean pitch of 6.51 feet.

The main condensers each have 5,382 square feet of cooling surface.

*Description of Turbines.*—The general arrangement adopted and the type of turbine employed provides two absolutely independent propelling systems, with suitable arrangements for reductions in the propelling power for cruising speeds. All the different ranges of speed are controlled by one maneuvering valve.

The Bréguet turbine is classed among the "drum and disc" compound impulse type. It utilizes the kinetic energy of the steam, created in the nozzles, entirely on the moving vanes. They are designed to meet the demands for variable speeds, from that of cruising to the maximum, in a manner as simple as obtained in the ordinary reciprocating engine.

There is one ahead and one backing turbine on each rotor, contained within the same casing. The thrust bearings are built in with the casing and are placed forward.

Between the two faces of each stationary disc containing the nozzles or guide vanes, exists a difference in pressure, upon which difference depends the velocity of the steam which passes through. On the two faces of each receiving or moving disc equal pressures exist.

Constructive features employed in the Bréguet turbine result in the following:

1. Considerable clearance between moving and fixed parts. This clearance is big enough so that in case of breakage of any vane (which, by the way, it is claimed, has not yet occurred), the broken piece will render only minimum damage.

2. For a given steam velocity the same power may be obtained with smaller tangential vane velocity than with the reaction type of turbine.

Theoretically, the Bréguet turbine is one with total admission; yet, in the marine turbine, in order to give the power corresponding to different speeds, partial admission is employed in the first disc, the effect of which is as follows:

1. To suppress the flow of the steam until reached by the first receptive disc.

2. To diminish in a considerable degree the steam flow for reduced power of turbine as compared with that of normal power. In the discs following the first the steam flows with total admission. The losses due to friction in the vanes of the steam are thus reduced.

In the turbines of the *Bisson* and *Renaudin* there are two steam inlets, while in their predecessors, with the Bréguet type of turbine, there were three inlets. Diminution in power results in greater steam expansion, just as in an expansion engine.

The turbine casing is divided in two parts through a horizontal steamtight joint. In this casing are fastened the discs which contain the guide vanes or distributors. These vanes are inclined in a way to direct the steam on the movable vanes after having been partly expanded.

On the extreme ends are rotor and thrust bearings. On the casing are placed the steam inlet nozzles, the exhaust nozzle and valve connections.

Around the shaft in the turbine ends are bronze packing rings which bring inside and outside pressures into a state of equilibrium. Labyrinth packings are provided at each end.

The rotor shaft bearings, of a special bronze, are made espe-

cially with a view to prevent the spilling or leakage of oil. The disposition of the oil grooves, together with the throwers on the journals, tend to repel the oil to the inside.

The rotor is made up of shroud rings, the moving (receptive) vanes, wrought-steel collars and the drum to which these collars are keyed. The rotor drum and rings are made of wrought steel, and are finished on the entire surface, in a manner such as to render same balanced for both static and dynamic forces, whereby vibration and strains incidental thereto are largely eliminated.

All vanes are made of high-tensile bronze, and are produced by a method of milling to shape in the lengthwise direction of the bar, which is afterwards cut in suitable lengths. Bronze is being used for vanes owing to less oxidation taking place with this than with ferrous metals.

The blades are fixed to rotor and stator as follows: The shape of the blade is practically like that of the DeLaval turbine blade. In the collars fastened to the drum and rings fastened to the casing the bulb-shaped blade root is inserted in slots of identical shape and then riveted over. The shroud ring is formed by slugs of metal being cast with the blade, all blades being securely tied together at the free end by a dovetailed copper ring hammered into a corresponding groove. The solidity of this arrangement may easily be conceived and its principal details have proved their worth in the DeLaval turbines where centrifugal forces are of great moment. Fig. 2.

The components in an axial direction, arising from the steam flow in the turbine, are practically zero, as is also the pressure difference on each side of the rotor discs. This is, however, not the condition in the steam-admission compartment, the static pressure difference in which is considerable and must be compensated for by pressure in the thrust bearings and by an arrangement of balance pistons. The pressures in the thrust bearings vary with the power developed, as part of it is composed largely of that caused by the propeller thrust.

*General Data.*—Forced lubrication is employed in all tur-

bine bearings, the apparatus being arranged in double units to insure safety. The main turbines exhaust each into an independent condenser. On the *Renaudin* there is being in use for the first time in the French Navy a Bréguet ejector, which is really a dry-air pump without any motive machinery whatsoever. This arrangement insures simplicity, safety and continuous working, which should prove of great value. Besides the foregoing, an apparatus like this offers reduced complication in mechanism and also in weight, together with, it is claimed, steam economy.

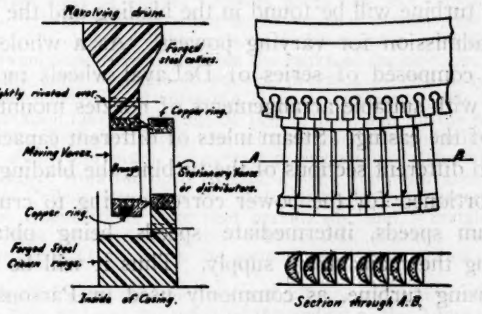



FIG. 2.

Electric current is produced by two dynamos of 12 kw., operated by petrol engines. The steam-generating plant consists of four Guyot boilers built by l'etablissement d'Indret. The following data have reference to the boilers:

| | |
|---|---|
| Heating surface, square feet | 5,172 |
| Oil burners | 11 du Temple, each boiler. |
| Maximum fuel consumption allowed, pounds per square foot of heating surface | 1.33 |
| Pressure, pounds per square inch | 228 |

The boilers work always under some degree of forced-draft, the maximum of which does not exceed 7 inches of water. There are placed in each fireroom two steam-driven fans, each of a capacity of 9,000 cubic feet per minute.

The fuel-oil tanks have a combined capacity of 180 tons, the theoretical steaming radius being 1,200 miles at 14 knots.

This type of destroyer carries two boats of 23 feet length and two 12-foot canvas boats. Anchor and chain equipment consists of two bow anchors, each weighing 1,825 pounds, and 160 fathoms of 1-inch chain; also three kedge anchors, each weighing 580 pounds.

The two pumps connecting to the main drainage system are capable of discharging 20 tons of water per hour.

*Comments.*—The distinguishing features in the Bréguet type of turbine will be found in the blading and the system of steam admission for varying powers. As a whole this turbine is composed of series of DeLaval wheels mounted on drums, with suitable arrangements of nozzles mounted on the inside of the casing. Steam inlets of different capacity supply steam to different sections of the turbine, the blading in which is proportioned for the power corresponding to cruising and maximum speeds, intermediate speeds being obtained by throttling the main steam supply. Thus it will be seen that the cruising turbine, as commonly used in Parsons' turbine installations, in the Bréguet turbine is incorporated within the same casing as that of the main turbine.

The steam admission for low power is only a partial one around the blade element, obtained by placing a sufficient number of nozzles in the distributing ring, differing in this respect from the Curtis system, where the steam is regulated by a number of nozzle valves.

The most valuable feature about this turbine, however, seems to rest with the blading, which, by the method employed in fastening, may, comparatively speaking, be considered "*safe.*" The sketch attached and description given in the text explain the construction sufficiently. The mechanical advantage of this method cannot be questioned and has been well proved in the numerous DeLaval turbines running at unusual speeds of revolutions and with stresses several times those thought advisable in other types of turbines, with hardly ever a case of blade stripping having occurred.

# NOTES.

## ENGINEERS AND ENGINEERING IN THE ROYAL NAVY.

In considering the subject of Engineers and Engineering in the Royal Navy the first to refer to is naturally the commissioned engineers. This most important class of officers numbers about 800, and consists of one engineer-in-chief, 12 engineer admirals, 19 engineer captains, 268 engineer commanders, 519 engineer lieutenants. Of this number it is pleasing to know that twenty-one have risen from the ranks of engine-room artificers. Regarding the entry of engineer officers, previous to 1903 a system obtained by which boys could enter the Navy as engineer cadets by means of an examination taken at the age of 14½ to 16½, and for the purpose of training cadets Keyham College was set apart, and the potential engineer studied for about four years, his parents paying an annual fee of £40 plus expenses. Apart from the open method, a most important avenue of entry into this College was from the ranks of dockyard apprentices. For this purpose examinations were held periodically, and a certain number of the top boys admitted at greatly reduced fees. To this system must be given the credit of placing in the premier service some of our foremost engineers.

The method of entry for commissioned officers which now obtains was initiated in 1903, and was the result of the finding of the Admiral Douglas Committee. It is a system of common entry; that is to say, all officers, whether deck, Royal marine or engineer, are entered by one common channel. For the purpose of entering cadets a Committee of Selection, presided over by the Secretary to the Admiralty, was formed, and before this committee appear the candidates between the ages of 13 years and 4 months and 13 years and 8 months. If selected by the committee as suitable they are set an examination, and on passing proceed to one of the training colleges. Osborne and Dartmouth are set apart and specially equipped for the training of these cadets. In each of the colleges the cadets spend two years, the training fees being £75 per year, and in addition the expense of books, clothes, washing, etc., are estimated at a further £40 per year. After leaving Dartmouth College the cadet joins a special training cruiser for about twelve months, and can then be made a midshipman and drafted to one or the other of the battleships of the Home Fleet. From the time he joins the training cruiser to the time he obtains the rank of sub-lieutenant, which would be about five years, his parents must make him an allowance of £50 per year, the minimum total cost to the parents of a naval officer being about £800.

During the period which has occurred between the stopping of the old scheme in 1903, and the useful results obtained from the new, a serious shortage of officers has occurred, to overcome which the Admiralty has decided to immediately grant lieutenancies to one hundred Naval Reserve officers, and they have also brought into being a temporary scheme of entry for cadets. A limited number of special cadets are to be admitted, at an age of about eighteen, to an open competitive examination, and those selected will enter the Navy on similar lines to the younger cadet. In addition to the above scheme it is intended to grant a certain number of commissions to men on the lower deck.

The duties carried out by the present class of commissioned engineers may be summarized as follows: The engineers-in-chief and one or more of the engineer-admirals are attached to the Admiralty at Whitehall for various duties; other admirals are attached to dockyards in charge of the engineering departments and for general staff duties. The engineer-captains are also in charge of dockyard departments, fleets and repair ships. The engineer-commanders are placed in charge of the engineering departments of large battleships and cruisers, and engineer-lieutenants are sent to ships of all classes, including torpedo destroyers and submarines. In addition to the above the commissioned engineers stand by ships being constructed, and carry out the important duty of inspecting, testing and passing work produced in the various shipbuilding yards.

The pay of engineer officers is as follows: Engineer rear admirals receive £3 per day, and engineer captains £1 15s. 0d. to £2, engineer commanders £1 4s. 0d. to £1 13s. 0d., and engineer-lieutenants 10/- to £1. Ranking after the engineer-lieutenants are the commissioned warrant engineers and the warrant engineers, and they may be called the practical engineers of the Navy, numbering about 670 and all promoted from the ranks of the engine-room artificer class. The duties of the warrant engineers are multifarious, and of late years, owing to the dearth of engineer-lieutenants, they have been placed in positions of much more responsibility than hitherto, and in a very large number of cases have taken the place of engineer-lieutenants. To understand the duties of warrant engineers it is first necessary to study the number of officers carried on a battleship. During the maneuvers of last year one ship carried one engineer commander, one engineer-lieutenant, and five warrant engineers, the work being allotted by the engineer-lieutenant between the warrant officers, one having the main engines, one the boilers, one the engine-room and stokehold auxilaries, another having the double bottoms, flooding arrangements and steam boats, and the other together with the engineer-lieutenant, exercising a general control over the whole department. By that means each officer specializes in one of these branches, and when the ship is not steaming he superintends all repairs in that particular department. When under way the watches are kept by one of the warrant engineers or the engineer-lieutenants (if more than one is carried), and he is responsible for everything which goes on in the whole engine-room department. Four watches are generally kept, that is, four hours on and twelve off. In addition to watch-keeping he must take what is known as day duty, that is, every fourth day his name is put up as engineer officer of the day, and he is responsible for everything which goes on in the whole department; he will take morning divisions, evening quarters, and go the rounds, also see the hands turned to at 6 A. M., and the engine-room artificers started on repair work at 7:30 A. M. In going the rounds it is usual to take an engine-room artificer, a stoker petty officer and two stokers. Every department is visited and the following particulars taken: water in main and reserve feed tanks, steam pressure, density, alkalinity, depth of water in bilges, temperature of cold room, and any other special particulars, these being reported to the engineer-commander and "all correct" given to the upper deck officer going the rounds. The engineer officer must be prepared to be called upon at any time of the night in case of breakdown of any of the innumerable auxiliaries. Another important duty of the engineer officer is coaling, and the time taken may be from ten to twenty hours; he must direct the stowing of the coal, and, owing to the great amount of trimming, due to the number of bunkers, a stoppage may occur at any time. Other duties include kit inspection, inspection of mess places, bath rooms and hammocks. It is also necessary to become acquainted with the stations of men, during fire quarters, collision quarters, man and arm ship, and general quarters, any of which may be called day or night.

Referring now to the engine-room artificer class. The age of entry is from twenty-one to twenty-eight, the minimum height 5 feet 4 inches, and engine fitters, coppersmiths, boilermakers and engine smiths are enrolled. It is first necessary to pass a medical examination, after which proof must be given of having served at least five years at his trade; should he satisfy the authorities, he is sent to one of the three naval ports, and examined in theory and practice. The theory consists of reading, writing and arithmetic, simple and compound, also knowledge of engines and boilers; he must understand and be acquainted with the names and uses of all engine parts, understand the uses of steam, water and vacuum gages, boiler mountings, how to take the density, alkalinity and height of water in the boiler, also how to deal with hot bearings and general breakdowns. The trade examination consists of fitting and turning, a favorite job being a butt-end strap for brasses with jib and cottar, and a piece of screw cutting. At the present time there appears to be a shortage of engine-room artificers, owing chiefly to the discontent and dissatisfaction caused by the introduction of the stoker mechanician scheme. The terms of service is for twelve years and the pay is, under three years' service 38/6 per week, after three, 40/3, after seven, 42/6, after twelve, 45/6, with 1d. per day for good-conduct stripes which are given, but not worn. After an engine-room artificer has served six years he is eligible to sit for the rank of chief engine-room artificer. This examination is very similar to the second-class Board of Trade certificate and carries with it, if under six years' service pay of 49/- per week and if over, 52/6. An engine-room artificer can earn a pension of £1 6s. 0d. per week after twenty-two years' service. All engine-room artificers rank immediately on joining as chief petty officers and enjoy the privileges of that rank. It might be mentioned that any engineer holding a first or second-class Board of Trade certificate, and who is below the age of twenty-eight, can enter without examination, provided he passes the medical examination. In addition to these methods of joining the Navy, there is another, viz., by joining as a boy artificer. Each year a limited number of vacancies is advertised, open to the sons of British subjects between the ages of fifteen and sixteen. There are three systems of entry. (1) By open competition. (2) By recommendation of naval commanders-in-chief and certain other officials, on account of service claims. (3) By competition limited to candidates recommended by certain educational authorities. The open competition first mentioned is held in April each year and is conducted by the Civil Service Commissioners in the same manner as for dockyard apprentices. All applications must be made to the Secretary, Civil Service Commission, and the examination fee is 2/6; the subjects taken are arithmetic for which 250 marks are given, mathematics 300, English 300, history and geography 250, science 300, total 1,400. The second method of entry is limited to candidates with service claims. These entrants do not exceed twenty in number. They are sons of warrant officers and chief and first-class petty officers and the examination is similar to that in the open competition. The third method is limited to candidates recommended by educational authorities; the candidates must have attended a secondary or higher grade school for at least twelve months. The examination takes place in October and lasts one day, the subjects being practical mathematics and elementary science.

On successfully passing the examination the boy artificer receives a free outfit and is sent to one of the special training ships. There he is educated and instructed in general engineering work for three months, at the end of which, one of three branches of practical engineering is chosen for him, these being fitting and turning, boiler making and smithing and coppersmithing and molding; he then specializes in that branch of the trade. During the first year he receives 6d. per day, second year 7d. per day, third year 8d. per day, fourth year 9d. per day; on completing the

fourth year he can qualify as fifth class and receive 3/- per day, after which he can pass for fourth class, and then be in every respect the same as an ordinary engine-room artificer. All leave granted to engine-room artificers is on full pay; only half the ordinary railway fares must be paid if an artificer wishes to proceed home.

*Messing.*—The commissioned officers mess in the ward room at a cost of about 1/6 per day; the commissioned warrant and warrant engineers mess with all other warrant officers in the gun room, or in a special warrant officers' mess, the cost of messing being from 9d. to 1/- per day. The engine-room artificers have a separate mess place on the lower deck, and mess entirely by themselves, this being one of the privileges of the rank, their mess account being about 15/- per month. The above sums are in addition to the service allowance of 8½d. per day. For the purpose of preparing and serving meals a certain number of stewards, cooks and domestics are allotted in proportion to the number in the mess, the engine-room artificers having stokers told off for a similar purpose.

*Sleeping Accommodation.*—Officers are allowed a cabin in order of seniority, the senior engineer-lieutenant generally takes one which is close to the engine room. In very few ships, however, are there enough cabins for all. The accommodation for the warrant engineers is often the same as for engine-room artificers, and consists of two hooks, by means of which a hammock can be suspended. In this respect the comforts of the engine-room staff compare very unfavorably with that of the engineers of the merchant service, and the life is thus made less attractive than it otherwise would be. The engine-room artificer employs a stoker (of whom there are plenty willing to do the work) to hang up and take down his hammock, for which, and also for washing it, 5/- per month is paid.

Perhaps a more detailed account of the duties of the engineers will be interesting. It is unnecessary to say that on joining a ship it is first desirable to get a general idea of the run of the vessel, and this is not so simple as it might seem, owing to the many different compartments and watertight doors. On the day of joining it is necessary to report to the captain or commander and then to the engineer-commander, and as soon as possible obtain sketches of the general run of steam pipes, main and auxiliary, boiler valves, sectional valves, bulkhead and other valves of equal importance, and the lead of the suction and discharge pipes of the various pumps. It is also necessary to know exactly the function of each pump, and for what purpose it may be used. The same applies to voice tubes, engine-room telegraphs, watertight doors, flooding arrangements, ventilation trunks, steering gear, etc., and it is only by obtaining a thorough knowledge of all these particulars that a feeling of confidence and authority can be obtained. It is always better to trace and find out things for oneself than to rely on information obtained from subordinates.

To give some idea of the duties of engineer officers we will take for example one of the cruiser class of vessels commissioned for the maneuvers of last year. The engine-room staff consisted of one engineer-commander, one senior engineer-lieutenant, and five warrant engineers, in addition to three chief engine-room artificers, twenty-three engine-room artificers, ten chief stokers, twenty-one stoker petty officers, twenty-seven leading stokers and 190 stokers; of this staff not one had ever been to sea in the ship. The engines were in two sets, having cylinders 36 inches, 59 inches and two 68 inches, with a stroke of 48 inches, indicating at 120 revolutions per minute 21,000 H.P. The boilers were thirty in number, of the Belleville type, and placed in four stokeholds; the auxiliaries were two main and auxiliary air pumps, two main and auxiliary circulators, two steering engines, two drain pumps, five air compressors, two hydraulic engines, four $CO_2$ machines, two refrigerating engines, four fire and bilge pumps, two capstan engines, two boat hoists, two blowing en-

gines, eight boiler-feed pumps, two hotwell pumps, twelve fan engines, in addition to the machinery of the workshop and steam boats.

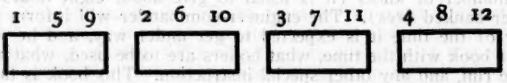
We will suppose that the ship is to proceed to sea at a certain time, the order will be given to the engineer-commander to have steam for a certain number of knots (it is usual to give about eight hours' notice if not under banked fires). The engineer-commander will inform the senior engineer of the time it is expected to get under way, and he will fill in the order book with the time, what boilers are to be used, what auxiliaries are to be run, and any other special instruction. This book is taken to all officers concerned and initialled by them, and about six hours before the ship has to proceed the engineer officer who is taking the first watch, will go below and see the following sequence of duties carried out. During the first hour he would inspect the boilers, and see the glasses showing water up to working level, he would also try the test cocks, open the air cocks and then have the bottom fires lighted. He would then see all gage cocks open and also all drains on main and auxiliary steam pipes and valves, he would have opened all boiler stop valves (sectional valves, bulkhead valves, intermediate screw-down valves), starting and maneuvering valves, also valves to steering engines, main circulating engines, starting engines, to sirens or whistles and drain pumps. The silent blow-off would be closed, and the jacket drains opened to the bilge, also the cylinder and slide-valve casing drains. The exhaust valves on the auxiliary condensers and the sluice valves of main and auxiliary circulating pumps would be opened, the air cocks on the condensers opened until the condenser is full of water. When fresh water accumulated in the condenser it would be tested for freshness to ascertain whether all the condenser tubes are tight. All crank pits and working parts of the main engines would be examined for obstructions, and the turning gear disconnected, the main feed tanks examined and the suction valves on the main and auxiliary feed pumps opened, the indicator and revolution counter connected and adjusted, the stern gland slackened back and funnel guys adjusted. All these duties would take the watch the greater part of an hour. In the second hour the wing fires would be lighted, and directly steam was showing in the boiler, the safety valve and water gage cocks would be tested, the regulating and maneuvering valve would be closed and the cylinders warmed by means of the jackets. As soon as possible the drain pump would be started and also the main circulators.

During the next hour the bottom fires would be spread, after which the furnace doors would be closed and the ash-pit doors opened, the cylinder drains connected and the condenser and air pumps started. At the end of the fourth hour the officer of the watch and all hands would be relieved, everybody in the watch waiting for his own relief, and not leaving until his own relief turns up. During the next watch steam would be raised to the pressure required, the sirens, whistle, voice tubes, engine-room telegraph and steering gear would be tried, and during the last hour of the watch the links would be run over and the main engines tried, permission being first obtained from the bridge.

Having got fairly under way, things in the general course of events settle down to easy working, but, owing to the constant changing of speed, due to the fact that the ship may be in line or performing some evolution, constant attention must be paid to the steam pressure. To relieve any excess of pressure and to prevent safety valves from blowing off, a silent blow-off is fitted. This is simply a by-pass valve leading from the main steam pipe to the main condenser, and can be opened to relieve the pressure. It is only used in emergency, as a rush of live steam into the condenser, even when a baffle is fitted, may have a tendency to damage the tubes.

To regulate the firing of a boiler it is usual to fit an electrical device

called a Kilroy firing indicator. This is fitted in the engine room, and can be adjusted to ring a bell in any stokehold every given number of minutes. Each furnace is numbered in this order:




and only one furnace is fired at a time, two men working together, one putting on the coal, the other opening and shutting the door. The plan adopted for cleaning fires is to do numbers 1, 2, 3 and 4 in one watch, and 5, 6, 7 and 8 in the next, and so on. The cleaning of tubes of water-tube boilers depends upon the quality of the coal used and the amount the fires are forced. The tubes of large-tube boilers are cleaned by means of a jet of steam or air discharged from a lance, and is generally carried out just before cleaning fires. Steam cleaning of small-tube boilers is only done in case of emergency.

*Stokers.*—It may not be out of place to refer to the stokers of the Navy. A lad joins as a second-class stoker, and passes through the grades of first-class stoker, acting stoker, petty officer to the rank of chief stoker. There is also a rating of yeoman of stores, and one of engineer-writer, and the rating known as mechanician can also be reached. It is this rating which has given rise to so much discontent amongst the engine-room artificers.

*Electricians.*—Until within the last eight years all electrical work, such as running dynamos, wiring, searchlights and electrical work, was carried out by engine-room artificers, and it was usual to give a certain number of them a three months' course of instruction in the theory of electricity and then draft one to each ship. This has now been changed, and the lighting plants come entirely under the charge of an upper-deck lieutenant, generally the torpedo lieutenant, under whose order the electricians work. The uniform, pay and promotion of the electrician are identically the same as those of the engine-room artificer, although the electricians mess with the chief P.O.'s and not with the engine-room artificers.

*Boilers.*—The marine or cylindrical boiler is seldom seen in the Navy, but they are still to be found in despatch boats and tugs. The water-tube boilers are invariably used, owing to their lightness and ability to carry a high pressure and to the speed with which it is possible to raise steam. Boilers in which the internal diameter of the tubes is greater than 1½ inches are classed as large-tube boilers, those with tubes below that size are classed small-tube boilers. Examples of the former are: Babcock & Wilcox, Niclausse, Durr; of the latter, Thornycroft, White Forster, Reed, Normand. The Yarrow boilers are made with both large and small tubes.

*Raising Ashes.*—Various arrangements have been devised for removing ashes and clinkers from the stokehold, and ash ejectors are fitted to all the latest types of ship. The type, known as See's ash ejector, was found to work most successfully, but it left a trail of ashes in the wake of the ship, and to obviate that a later form of ejector is used in which the ashes and water are discharged through a pipe in the ship's bottom.

*Feed Pumps.*—In nearly every case the feed pumps are of the Weir type.

*Watertight Doors.*—To allow for communication, doors are cut where necessary in bulkheads and decks dividing the various watertight spaces. These doors are of two distinct types, sliding and hinged. In all main transverse and longitudinal bulkheads, sliding watertight doors are fitted.

Where the coal bunkers lead into the stokehold vertical doors are fitted, in the majority of other cases horizontal doors are fitted. Every watertight door is divided into one of three classes and marked A, B, C. "A" doors are closed when the order "close watertight doors" is given; "B" doors are kept closed when maneuvering or when entering or leaving harbor, at night, or in a fog; "C" doors are always kept closed when at sea.

*Pumping and Draining.*—The main circulators are arranged to take their suctions if necessary from the engine-room bilge, and a main drain of about 16 feet in diameter is run from the after end of the forward boiler room to the forward engine room, a branch being taken to each engine room. By this means water can be carried from any stokehold bilge to either engine room. It is usual to fit a sluice valve and also a non-return valve in each stokehold. A sluice valve is also fitted between the two engine rooms. For dealing with small quantities of water in the bilges, etc., suction pumps are taken to the fire and bilge pumps. These pumps and also the Downton pumps, have connections to the main suction pipe. This is a pipe about 6 inches in diameter running the full length of the ship. Branches from this pipe are led to nearly every compartment in the hold and double bottoms, and each branch is fitted with a valve box which can be used as a non-return screw-down and flooding valve. The magazines are not fitted with a branch to the main suction, and if it is necessary to pump out a magazine a hose pipe would be rigged to the nearest Downton. In the latest and most modern ships the main drain has been dispensed with, and the main circulators can only take their suction from the engine-room bilges. The main fire and bilge pumps are arranged to pump from the engine-room space or double bottom directly under it. Arrangements have been made to pump out each boiler room and the spaces under it with a special pump. Outside the engine and boiler rooms motor-driven pumps of the centrifugal type have been fitted, which can be made to pump out the spaces in the compartments.

*The Royal Naval Reserve.*—Previous to the year 1904 it was possible for any mercantile marine engineer to become a commissioned officer in the Royal Naval Reserve. The fact of holding a Board of Trade certificate was considered sufficient qualification for his entry. Since that date entry as commissioned engineers has been entirely stopped and a new system substituted. Under this new scheme a few specially qualified engineers, holding first-class Board of Trade certificates and being under forty years of age, are to be admitted as warrant engineers provided they are recommended by their employers as having a thorough practical knowledge of the machinery of His Majesty's ships. The position of a warrant engineer is identically the same as that of an artificer engineer, with whom he ranks, but as this applies also to the warrant mechanician it does not count for very much in the eyes of the Admiralty. Cases are known where warrant engineers have been recommended for promotion by officers of as high a rank as rear admiral, and it has been refused, even though they have offered to give up their retainers. Apart from the question of mechanicians, the cabin accommodation will alone stop the best men from joining the Reserve as warrant engineers. When embarked for training or the maneuvers it is not uncommon to find the admiral's steward or valet allotted a cabin and the warrant engineer provided with a hammock, and this when working in three watches.

Whether the new officer scheme of entry has anything to do with the closing of the commissioned engineers' lists is a moot question, but, as previously suggested in a former issue, some form of examination might be held for mercantile engineers to obtain either direct entry as commissioned officers or be ranked from the lists of warrant engineers.—"The Marine Engineer and Naval Architect."

## WHAT'S WRONG WITH GERMAN SHIPBUILDING?

From time to time one or another prominent person raises the question in Germany as to why that country does not secure more orders for ships from foreign owners, and each man voices his own particular theory. We are assured by Dr. Franz Hochstetter that there are too many German shipbuilding yards, in view of the comparatively small number of German shipping enterprises, so that there are not sufficient orders from these to go round and far too few orders from foreign owners to meet the deficiency. He attributes the precarious position of the German yards very largely to British competition abroad, which has so far succeeded in capturing the vast majority of all the foreign orders that are going. It is now the turn of no less a personage than Herr Ballin himself, who, so far as the scarcity of shipbuilding orders from abroad is concerned, lays the whole blame at the door of the German Government. In an article in the Berlin "Lokalanzeiger" he attacks Germany's foreign policy, and particularly the unwillingness of Germany to participate in large foreign loans, unless in conjunction with some other Power. It is not enough, he says, that Germany should husband her own resources by the provision of capital to nurture her home industries. He points out that Great Britain and France take their money abroad with great political and, industrially, practical advantage, hoping that part of the money which they lend shall return in the form of orders, though the cry at present, among British manufacturers like Mr. Docker, is that the position is just reversed. The German banks, who finance undertakings in Argentina or South Africa or Asia Minor, or elsewhere, take good care that nearly all the money comes back to Germany in orders. "Just look at German shipbuilding," says Herr Ballin, however; "there the consequences of our foreign money policy are brutally evident. Today, without any doubt, we are, as regards the quality of the work of our shipbuilding yards, on a level with the very best British performance. Our prices are cheaper and our contract dates for delivery are more trustworthy, since, in consequence of the bad conditions among the British workmen, large British shipbuilding yards will give, only with the greatest reluctance, any definite date for delivery at all. Nevertheless, with isolated exceptions, the orders of the whole world go to Great Britain."

Too much importance should not be attached to the comparison between the ships of the respective countries. No one is indebted to the shipbuilders of this country more than Herr Ballin. In fact, he is still having boats built over here—we recorded the launch of the *Holsatia* last week and the trial trip of the *Frisia* this week—so that evidently the price, the delivery and even the quality come up to his requirements. It is, however, Herr Ballin's belief, as expressed in the "Lokalenzeiger," that the sooner a policy of foreign loans repaid by orders is entered upon the sooner foreign orders for ships will begin to be placed in Germany. The difficulty is that Germany needs most of its money for industrial developments at home.

But if the German Government is really lacking in this particular spirit of enterprise and is loath to start on such a money-lending policy, it cannot be said that as an advertising agent it has not dealt most generously with German industries, and especially with the German shipbuilding industry. The most recent example is the journey which Prince Henry of Prussia made in the *Cap Trafalgar* to South America—a country on which Germany keeps its eye steadily, especially for shipbuilding orders. The Prince himself has admitted in his speeches over there that if his journey was not primarily for using his influence in favor of German industry, and especially German shipbuilding, it could not fail to have vast economic possibilities. There is nothing to be said against such a policy, and over

and over again the opinion has been expressed in the British Press that it would be well if Great Britain's State representatives abroad used their influence a little more to further British industry. But it is against the grain of the British statesman, even against the British spirit of independence in general, to do so; and is British enterprise much the worse for it?

There can be no doubt, however, that the shipbuilding industry in Germany is not in a very flourishing condition. While it suffers almost as much as we do in this country from industrial disturbances, it does not receive the same relative number of orders, and the orders and the prices which it does receive are quite insufficient to enable a good return to be earned upon the capital spent on German yards. With some of the technical reasons for this state of affairs we have already dealt; but it would appear that the German shipbuilders went ahead rather fast in spending capital at the outset before they had built up their business. With cheap labor, cheap steel and modern yards, the Germans ought to be able to build cheaply, even although for the highest class and largest size of vessel they have not yet the same experience as this country. They study our own development so carefully and have had so much help from men trained in British yards that it is only to be expected that they should in time be able to turn out even the most difficult vessels. Sooner or later the immense amount of thought and study which is being given to naval architecture in Germany is bound to have its effect upon the science generally. But even when Germany is able to produce ships of all kinds as good as those produced in this country—and in certain directions it is doing this already—there still remains the need for obtaining orders. At the present time not only is German shipbuilding suffering from the fact that there are not enough orders from German lines to go round, but it is also hampered by the fact that the names of its builders are so little known in other parts of the world. It is all very well to send vessels like the *Von der Tann* and the *Cap Trafalgar* to South America, both of which are fine examples of German shipbuilding. The number of orders for vessels of this class is very limited, and what is wanted is that the ordinary German shipbuilders should get their names and their products known to shipowners throughout the world, and this can only be done by constantly advertising their work. The advantages of publicity have long been appreciated in Great Britain and in the United States, as the advertisement columns of "Shipbuilding and Shipping Record" and its contemporaries show, and the fact that publicity is not sufficiently appreciated at the present time on the Continent is undoubtedly something of a handicap in enabling German shipbuilders to get foreign orders. Now-a-days one must keep one's name to the fore.

If the German shipbuilding industry relied more on itself and its resources instead of leaving too much to the Government and its agents, it would, in our opinion, fare better in the markets of the world. It leaves too much to the Government—it does too little itself. But every nation has its own special qualities, and the Germans love being governed and organized as much as the Briton loves being let alone and allowed to "muddle through."—"Shipbuilding and Shipping Record."

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## MODERN GERMAN WARSHIP DESIGN.

"The Engineer," February 27, 1914.—An article discussing the general trend of German warship design illustrating successive types of battleships and cruisers and comparing various features with those of contemporary vessels.

This article is instructive as representing the essential features upon which German naval thought and experience have arrived at decisions.

## SOME QUESTIONS RELATING TO BATTLESHIP DESIGN.

By T. G. Owens. Paper read before the Institution of Naval Architects, reprinted in "The Engineer," April 3 and 10, 1914.—This paper deals extensively with features of battleship design, especially armament and protection, will be of special interest to naval officers. This paper is extensively illustrated with complete diagrams showing all the principal features of the designs proposed.

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## HULL EFFICIENCY.

Propulsive efficiency is composed mainly of three factors, known as the three E's, viz: engine efficiency, propeller efficiency and hull efficiency, and when the effective horsepower includes the resistance of appendages, such as bilge keels, bossings, &c., these three are the only factors in the case. Generally, however, the ratio $\frac{\text{E.H.P.}}{\text{I.H.P. or S.H.P.}}$ means the ratio of the effective horsepower (or the horsepower necessary to overcome the resistance) of the naked ship, *i.e.*, without bossings, &c., to the indicated horsepower at the cylinders, or to the shaft horsepower just abaft the turbines. Accordingly, the propulsive efficiency generally equals the above three E's multiplied by what has been termed the hull appendage factor. The latter makes allowance for the increase in resistance due to all protuberances.

The propulsive efficiency value is, on the average, about .5, but when the value in actual experience ranges from .42 to .62, or thereabouts, it is necessary, where accuracy is desired, to fix this coefficient with some degree of precision. This is accomplished by assigning to each factor in the propulsive efficiency a value which experience or experiment has shown to be near what may be expected in any new problem. Engine efficiency, for example, may be taken as .84 in slow-running reciprocating engines in which all the pumps are driven from the main engines, whilst the limits in the other direction are reached by the fast-running engines of cross-Channel steamers in which .9 may be taken. In turbine work it is generally assumed that about 3 per cent. of the power is lost between the point where the torsion is taken and the propeller, this having been absorbed in friction at bearings and stern-tube glands; accordingly .97 is the figure taken in such work. For the determination of the propeller efficiency reference is in general made to the experimental work of Froude or Taylor, and the efficiency of the experimental screw of a pitch ratio and a disc-area ratio, corresponding to the actual screw of the ship, is taken at the same real slip at which the large screw is giving the required thrust. When proper allowance has been made for the rougher surface of the actual screw as compared with the experimental, for the greater thickness of blades and for the greater diameter of hubs (such allowances *in cumulo* being in the average about 10 per cent.), a fair idea of the propeller efficiency may be obtained.

The allowance for the resistance of bossings, &c., which is made in what is termed above the hull-appendage factor, can also be determined with some degree of accuracy. In a paper read by the then Commander Dyson, of the United States Navy, to the American Society of Naval Engineers (see vol. xxii of "Transactions") is given the results of many experiments carried out by Taylor in the Washington experimental tank on various types of ships run, first without appendages and afterwards with.

In this paper certain coefficients are given for the allowances for rudder, shafts, struts, bilge keels, docking keels, bosses, swells, &c., and these are of great use in determining this hull-appendage factor.

The only other term in the propulsive coefficient is the hull efficiency, and this is probably the most elusive of the lot, and the one about which least is known. In general, the value taken is unity, but it has a range of at least from .9 to 1.1. It can readily be seen that if the latter figure be taken instead of the former when computing the propulsive efficiency, a considerable difference may be obtained on trial from the performance anticipated. The information procurable on the subject of hull efficiency has been given by four experimenters, viz: Froude, Pecoraro, Luke and Peabody, and certain most instructive information can be gathered from their work. A word of explanation as to the meaning of the term is desirable. Suppose the model of a vessel to be towed at a speed V, and the resistance procured to be R, with no propellers working behind the model. It is found that when model propellers (of the same proportion to the ship's screws as the model is to the ship) are made to revolve behind the model at such a rotary speed that the thrust they are giving equals the resistance of the model, that the latter is no longer what it was when being towed without propellers behind. It has been increased by a definite amount, and this augment to the resistance is termed the thrust deduction, having been caused by the propeller action. The cause of this increase is that the influence of the screws pervades a column of water of substantial dimensions in fore-and-aft direction, and this influence extends far enough forward to cause a marked diminution in pressure on the after part of the ship, thus causing a virtual increase in resistance. The propellers are called upon to exert a thrust equal to this augmented resistance.

Another factor, however, enters into the problem. It is found that besides the resistance being increased, the propellers are giving the required thrust at a fewer number of revolutions when working behind the model than when advancing at the same linear speed with no model in front. The wake following the model has thus some beneficial effect on the propeller performance.

Although this wake following the model be turbulent in character its effect on the screws is equivalent to that of a solid stream advancing at some definite speed. This is proved by the fact that if the propellers are made to revolve (with no model in front) when advancing at a particular speed $V_1$ (less than V by the wake speed), the same performance is obtained from the screws as when working behind the model at speed V. If thrust $\times$ velocity be a measure of the thrust horsepower, the less the $V_1$ is, the smaller will be the necessary power.

These two factors, viz: the thrust deduction and the wake, are brought into the following form. The resistance has been increased, and the increase is expressed as a fraction of the augmented resistance, thus $\frac{R_A - R}{R_A}$ = thrust deduction, or $t$; and the wake as a fraction of the speed of the ship corrected for wake, thus $\frac{V - V_1}{V_1}$ = wake fraction, or $w$. The effective horsepower is proportional to R V, and if R be divided by $(1 - t)$ this is equivalent to increasing the resistance to augmented resistance. Similarly with the wake fraction, the V is divided by $(1 + w)$, which has the effect of reducing the velocity by the speed of the wake. We have, accordingly, $\frac{RV}{(1 - t)\ (1 + w)}$, and the value $(1 + w)\ (1 - t)$ is called the hull efficiency. It is evident that the higher the hull efficiency is the smaller will be the thrust horsepower in relation to the effective horsepower, accordingly the more economical the propulsion.

The question then arises as the possibility of controlling the hull efficiency value. In no case, as far as the writer is aware, has the form of a vessel been designed with the object of obtaining a high hull efficiency value. All the investigations that have been made have been in the direction of finding with a given form the effect of speed, propeller position, diameter, pitch and surface of propellers on the hull efficiency, as also the effect of number of blades, number of screws, and, lastly, but most important of all, effect of angle of bossings and direction of rotation. The most complete set of experiments on wake and thrust deduction ever undertaken was carried out by Mr. W. J. Luke, who presented the results to the Institution of Naval Architects in 1910. He discovered that in a single-screw vessel a much higher hull efficiency value is obtained than in one propelled by twin screws, that hull efficiency decreases slightly with increase in speed, and varies only very slightly with change in pitch ratio. Proximity to the shell, he found, has a considerable influence on wake and thrust deduction, and it appears that propellers should be placed as close to the hull as practical considerations will allow. The direction of rotation has little effect on the hull efficiency in vessels fitted with struts (such as warships), but when full bossings are fitted the direction of rotation has a wonderful influence on the performance. In a vessel fitted with horizontal bossings, a hull efficiency value of about .95 would be obtained by inward-turning screws and 1.1 by outward-turning screws. This would probably mean in most vessels a difference in speed of from 1 to 2 knots per hour. In actual experience a certain vessel, the *Niagara II*, fitted with horizontal bossings, obtained 12.8 knots with 2,100 H.P. with inward-turning screws, whilst with the screws revolving outwards a speed of 14.12 knots was obtained with an expenditure of 1,950 H.P. As the slope of bossing is increased the difference in the hull efficiency obtained is minimized. In actual practice it is safest to make all the screws revolve outwards.

Major Pecoraro has found by experiment that the greater the number of propellers introduced for the propulsion of ships the smaller becomes the hull efficiency, that value being higher in a single-screw vessel than in a twin, and in one driven by three screws than in one driven by quadruple.

Prof. Peabody has experimented with a large model of a single-screw vessel and discovered that distance from propeller post to propeller has a considerable influence on performance, and that this distance should be as much as practicable.

Dr. R. E. Froude, in 1898, gave to the Institution of Naval Architects many results of experiments conducted by him at the experimental establishment of the British Admiralty situated at Haslar. These results refer almost entirely to warship work. His experiments showed that there was practically no difference in hull efficiency value through direction of rotation, but since no bossings were fitted on these models it would be unsafe to accept this as a general rule, since the later experiments previously referred to demonstrated the importance of direction of rotation with certain angles of bossings.

Although a great deal of information in a general way has thus been given by experimenters, yet the hull efficiency value in any particular case can only be arrived at by tank experiment. Those who do not possess such an establishment, however, may learn from the data already given the directions in which to go in order to obtain the greatest value from this interaction between propellers and ship, and to avoid the conditions that make for low hull efficiency.—"Shipbuilding and Shipping Record."

## METALS AND CORROSION.

### THE INFLUENCE OF NICKEL ON SOME COPPER-ALUMINUM ALLOYS.

By Prof. A. A. Read, M. Met., F.I.C., and R. H. Graves, M. Sc.

This paper, read before the Institute of Metals, March 18, 1914, is reprinted in March 20, 1914, number of "Engineering," is a very exhaustive description of this subject and is of special interest. Tests of different compositions and resultant effects are given in a very thorough and comprehensive manner. The article is fully illustrated with curves of performance and microphotographs of samples.

### THE FAILURE OF CONDENSER TUBES.*

By Sir Henry J. Oram, K.C.B., F.R.S., &c. (Engineer-in-Chief of the Fleet).

I feel somewhat guilty in accepting the presidency of the Institute of Metals, seeing that I am often busily engaged in endeavoring to dispense with non-ferrous metals in warships' machinery in cases where experience shows, as it often does, that the qualities of ferrous ones are more suitable.

New warship machinery of today is greatly different in this respect from that of some years ago, and when these newer ships come into the ship breaker's hands many years hence, there will not be found that wealth of copper, brass, gun-metal, &c., which used to be common in the machinery, and which added so largely to the scrap value of the older ships.

It will be interesting to note the records showing the amount of copper, brass, gun-metal and other non-ferrous metal work that a modern warship contains in her structure, that is, the hull, machinery, guns, mountings and armor, as compared with the amount of steel and iron.

Although the non-ferrous metals cannot usually compare with steel in structural strength, there are still many positions in which other qualities than strength are necessary, and these metals are still indispensable. Approximately every 100 tons of iron and steel used in modern warships requires to be supplemented by 6 tons of copper or its alloys in a battleship, or 8 tons in a cruiser, to give the completed warship her final and necessary qualities. The increase in the non-ferrous requirements of the cruiser is due to the smaller proportionate amount of armor, and the higher power of the machinery, which latter contains a larger proportionate amount of non-ferrous metals than the remainder of the structure.

The propelling machinery in the modern battleship referred to above contains a weight of non-ferrous metals equal to 17 per cent. of that of the contained steel and iron. If we go back twenty years and consider a battleship of that period, we find that the non-ferrous metals in the propelling machinery amounted to about 34 per cent. of the steel and iron, or just double the percentage at present obtaining.

In the hull structure, excluding the propelling machinery, the proportionate amount of non-ferrous metals has, however, not been reduced as compared with twenty years ago. Although steel has taken the place of copper and its alloys to a large extent, this reduction has been counterbalanced by the considerable increase in gun turrets and their machinery, fire control appliances, electric lighting, telephones, &c., where non-ferrous metals are requisite. Considering the hull structure alone, excluding propelling machinery, the figures for the modern battleship referred

*Institute of Metals, Presidential Address. Slightly condensed.

to are that the non-ferrous metals amount to 4.4 per cent. of the iron and steel, while for the battleship of twenty years ago the proportion was 4.2 per cent., thus showing a slight increase in non-ferrous metals.

Except for the very smallest pipes, copper steam pipes have not been used in new ships for many years, and one recalls the feeling of relief at the Admiralty when the last copper steam pipes containing high-pressure steam had been safely "wired" or replaced by steel before further loss of life occurred.

Gun-metal ends for condensers have also disappeared, and been succeeded by cast iron, to the great advantage of the condenser tubes, and the condenser shells have now also for some years been made of sheet steel, which is lighter, and, although not so durable as gun-metal, is sufficiently so having regard to the probable life of the ship. Feed-discharge pipes and fire-service pipes, made of copper for so many years, are also now made of steel.

Although, however, in the directions indicated the non-ferrous metals are giving place to steel in warships, the application of these metals in some other directions is steadily increasing, so that there is no material reduction in the area of usefulness which exists for non-ferrous metals, and in this area the need for further accurate knowledge and experiments to enable these metals to reliably fulfil the purpose for which they are fitted is perhaps greater than ever. It is the object of the Institute of Metals to develop and foster such experiments and to increase such knowledge, and I am sure the Institute will not be found lacking in this duty. There is still a field of usefulness for non-ferrous metals not yet closed, owing to the inability of makers to produce satisfactory steel castings. Complicated machinery castings are no doubt difficult to produce in steel, but whatever the reason may be, the delay and occasionally even the impossibility of at present obtaining them in cast steel have obliged the Admiralty to rely more largely on gun-metal for certain parts than the merits of that material appear to warrant.

In the event of a cast-steel valve box being found defective, there is often no option at present except to replace it in gun-metal, otherwise we should have warships delayed in delivery or withdrawn from active service for an undesirable period.

This being so, it is unfortunate that gun-metal loses at high temperature so much of its strength, and powers of elongation under stress before fracture, but we may console ourselves with the knowledge that under a steady load a large amount of elongation is not really essential, although desirable, and there is the fact that in the British fleet there are thousands of gun-metal valves which have been working at high temperatures and pressures for years, in which there have been no failures due to brittleness or less of strength at high temperatures.

What exactly occurs with gun-metal at high temperatures is at present not sufficiently established. In the last two or three years three different and quite independent experiments on this subject, carried out by competent workers, have come under Admiralty notice, which differed considerably in the results obtained both as regards the loss of ultimate strength and elongation before fracture.

One must conclude that considerations, such as the temperature of pouring, the exact method of mixture, the time the specimen is allowed to be under the influence of the heat, and other factors, bulk much more largely in influencing the results of such experiments than is usually admitted, and exercise so large an effect on the results as to render them unreliable as a guide to actual practice, unless fully considered and allowed for. The exact effects of temperature and such other influences, knowledge of which would be of much value, seem to require a lengthy and systematic series of experiments, which I believe have not yet been undertaken so thoroughly as the importance of the question warrants.

This Institute has taken up the question of corrosion of condenser tubes in great detail, having appointed a representative committee to deal with the matter. The second report to this committee by its investigator, which was presented and read at the Ghent meeting last year, is a most valuable document to those interested in the question, and both directly and indirectly it will well repay the time spent in its perusal and study. The report was of special interest to the Admiralty, as it confirms Admiralty practice and experience of the last twenty or thirty years.

On examining the last report of this committee it is of interest to note the good results given by a condenser-tube alloy containing 2 per cent. of lead. Hitherto engineers have had a considerable prejudice against lead in the copper-tin-zinc alloys, and perhaps the prejudice was not very definitely founded, but the satisfactory results given by the alloy referred to above, and those given by a gun-metal alloy containing a small percentage of lead, at high temperature, should cause the prejudice to come under review, and some light appears to be forthcoming on this important matter.

It is possible that the percentage of lead in such alloys must be confined within very narrow limits to give the physical qualities required, and that outside these limits the alloy is unsuitable for the purpose intended.

If this view be correct, and the proper percentage of lead, and the limits of variation from this percentage that can be allowed, can be determined in the case of alloys intended for certain purposes, such as for use with high-pressure steam, a great want will have been met.

In order that progress may be attained, and an improvement realized in the adaptability of a material to its designed end and practical use in engineering, close coöperation should exist between the scientist and designer, the maker, and the user, and such an institution as this, forming a center for the free discussion and circulation of opinions, should be of the greatest utility.

The number of papers read before this Institute dealing with the finished product on the practical side and from the user's point of view have been comparatively few, and on this account the remarks respecting condenser tubes which I propose to make may be of interest, for in this case it was only by such mutual coöperation and interchange of ideas that a real advance was obtained as regards increased reliability of these important items of a warship's machinery equipment. There is no part of the machinery of ships on which more thought and time has been bestowed, or which has a more far-reaching effect on the reliability of the machinery than that seemingly simple article, the condenser tube.

Although the corrosion of condenser tubes is a defect of a serious importance, many of the worst troubles that the Admiralty have experienced with such tubes have not been from corrosion or perforation, but from splitting when at work, which is a far more serious defect. The number of failures of tubes due to corrosion and perforation is much in excess of the number that fail due to splitting, but the more serious consequences which result from splitting are such as place these defects in the front rank of undesirable occurrences.

In the effort to avoid the cause of this defect it will have to be borne in mind that the tube has to be soft and ductile, so that it will resist any tendency to splitting or flying, but at the same time it must not be too soft to resist the pressure of the packing material which is applied at the ends to prevent leakage. As these two requirements are necessarily mutually antagonistic, difficulty is found in uniting them in condenser tubes.

One of the many orders issued by the Admiralty with the view of minimizing corrosion in condenser tubes is an instruction to ships' officers to drive the circulating pumps at high speed at frequent intervals to clear tubes of cinders, &c. Although the second report to the Corrosion Committee appears to throw doubt on the influence of such matter in starting

local corrosion, yet, as the report also points out, advantages from the corrosion point of view in keeping low condenser temperatures, and bearing in mind that any obstruction in the tube reduces the flow of water, and hence raises the temperature of the tube, the instruction is justified by this further consideration. In addition, a single line of flow for condensing water is pointed out as beneficial, and all these recommendations are worth bearing in mind and applying as far as practicable; but where high powers have to be squeezed into the minimum of space, as in warships, we cannot insist on such consideration, and are often thankful to be able to get reliable machinery at all in the space allowed.

As the condenser tube has bulked so largely in the report of the Institute's Corrosion Committee, I propose now to give a brief account of the development of Admiralty tests and conditions during the last twenty-five years, and have referred to a few of the opinions which the tube makers have been good enough to favor the Admiralty with, this information being given in the hope that it will be of interest to various makers in comparing notes of their competitors' opinions and also of advantage to the Institute generally.

Prior to 1890 the condenser tubes in the Navy were made of the 70-copper, 30-zinc composition, but, owing to repeated failures of condenser tubes in the fleet, the Admiralty specification was in that year revised, and the alloy altered by adding 1 per cent. of tin in lieu of 1 per cent. of zinc, the revised composition being 70 per cent. copper, 29 per cent. zinc, 1 per cent. tin. An effort also was made, even so long ago, to introduce a flattening test, but it was not embodied owing to opposition on the part of the makers, and instead heating and jarring tests were introduced.

In 1891 the quality of the copper used was specified to be not less than 99.3 per cent. pure, while a little later tubes which were too soft were guarded against by a clause specifying that they should be stiff enough to stand the packing being screwed up tight enough to hold 30 pounds' pressure of water.

The specification was next amended so as to require tubes drawn from billets to be accurately bored and turned before being subjected to the final draws. It was found that in making the shells from which tubes were drawn, the bore was frequently eccentric, and often excessively so, and therefore the finished tubes were often thin on one side, although the total weight per foot was correct.

The defect of splitting became very comm n with the development of torpedo-boat destroyer machinery, and in 190( the tube makers were asked several questions by the Admiralty, among them being: (1) Whether boring and turning the shells before drawing would facilitate the rejection of those which may produce tubes likely to develop splitting. (2) Whether any alteration of the alloy was considered to be desirable, or whether the splitting might be due to impurities introduced with either of the elements employed.

As regards (1), the replies indicated that some firms thought that boring might be advantageous, but the general opinion was against both boring and turning.

As regards (2), no alteration of the alloy was suggested, except by one firm, who thought 1 per cent. of tin was a disadvantage, and that an alloy of 2 copper to 1 zinc would render splitting less frequent; while another represented that splitting must not be attributed to impurities in the metal.

In 1901 some important tests were introduced; among others: (1) A flattening test to one-half the original diameter without previous annealing; (2) heating to a dull red heat without splitting; (3) selected tubes to be sawn through and examined for internal defects; (4) the tubes should be drawn on a mandril.

The flattening test proved to be the most important addition, and pro-

duced a great stir among the tube makers when the results became known, and rejections were common.

Representations were made to the Admiralty by the tube makers concerning the flattening test, with the result that after much consideration a letter was issued to the makers informing them that this test would not be enforced at present, as far as rejection was concerned, but would be continued to be made for Admiralty information.

In acknowledging receipt of this letter, however, one firm stated that they were strongly in favor of this test, "which we consider to be both proper and desirable in order to ensure freedom from splitting when the tubes are in use."

In 1902 the condenser-tube makers were again asked to suggest alterations to the specification with a view to obviating splitting, pitting, &c.

They were asked to remark, among other matters: (1) As regards any alteration proposed in the alloy; (2) on the tests then specified, and with regard to the flattening test then in abeyance, whether a bulging or a flattening test was preferred; (3) whether they could supply tubes having diameter within narrow gage limits.

As regards (1), firms either considered the specified alloy suitable, suggested mercantile marine practice (70:30 alloy), or (*a*) one firm suggested 60:39:1 alloy, and offered to make tubes of this alloy, but later withdrew the suggestion. They also suggested 1 to 2 per cent. of manganese in lieu of tin; (*b*) others suggested their various special tubes, but did not bring any proof that they would be superior to the Admiralty alloy.

As regards (3), the firms could work to a margin ranging from 0.003 inch to 0.01 inch.

With regard to (2), most firms preferred flattening to bulging, but one firm preferred neither, stating that flattening and bulging tests encouraged the making of tubes of a semi-soft nature and of unequal densities throughout.

The great majority of firms, after a short time, adopted more or less the Admiralty view, the use of electrolytic copper becoming more common and the purest qualities of zinc and tin commercially obtainable being used.

In 1904 there was another revision of the specification, after personal visits to the firms by Admiralty overseers. The following stipulations, amongst others, were introduced: (1) The amount turned off and bored out of billets for condenser tubes should be sufficient to remove all surface imperfections. (2) The external diameter of the tubes to have a limiting variation of 0.005 inch. (3) The flattening test was reintroduced, ⅝ inch tubes to be reduced to ½ inch. All tubes to be left 2 inches longer than the required length, so that every tube could be tested.

There was some opposition to these new provisions, the principal difficulty being as regards the boring and turning conditions, which, however, were retained.

A Machinery Designs Committee was at that time in existence at the Admiralty, consisting principally of engineers outside the naval service and they recommended that the Admiralty should define the extent of the final draw in the manufacture of condenser tubes, and a letter was addressed in 1905 to four of the principal tube makers asking their opinion on this matter. At the same time they were also asked their opinion as to the exclusion of "scrap."

In their replies the firms were all in favor of clean "scrap," and gave their list of draws. One expressed the opinion that the material used was far more important than the number of draws, and asked that the original flattening test should be reintroduced—*i.e.*, flattening to half the original diameter.

All the firms were again consulted in 1906 as regards the advisability of

specifying: (1) The draws of the last but one and last tubes, together with the dimensions at each stage. (2) The amount bored out of the castings to be not less than ¼ inch of diameter. (3) The copper used to assay not less than 99.7 per cent. pure. (4) Scrap not exceeding 20 per cent. may be used, providing it is strictly limited to the ends cut from tubes during manufacture. (5) The flattening test to be applied to either end of the tubes and the diameter of ⅝-inch tubes to be flattened to 7/16 inch.

The replies of the various firms were as follows:

Only one firm objected to the limitation of scrap. Three firms objected to the draws specified. Three others were prepared to accept all the tests and requirements. Two others considered the purity of the copper required was too strict, and suggested 99.5 per cent., while another considered 99.7 per cent. copper a mistake, as it would involve the use of electrolytic copper, which they deprecated.

One firm objected to the amount bored out, while another considered ¼ inch was too much, and that 1/16 inch would be sufficient; and, lastly, one objected to turning the castings, but agreed that boring was beneficial, and also agreed with the purity of the copper required, and were prepared to agree to the draws specified, but considered that variation of hardness would result.

After considering these replies it was decided not to specify anything concerning the drawing of the tubes, and a specification for condenser tubes was issued in 1906, which included the following: (1) The copper to assay not less than 99.6 per cent. (2) Scrap to be limited to ends removed from process tubes, and not to exceed 20 per cent. (3) Total impurities not to exceed 0.625 per cent. (4) The amount bored out of castings to be not less than ¼ inch of diameter.

This revised specification has held good to the present date, with the addition that electrolytic copper may be used, and its success, when coupled with some improvements in condenser construction, has well repaid the time devoted to this important matter.

In 1908–9 many cases of split tubes occurred in a new battleship's condensers, and a suggestion was made to the Admiralty that a tensile test should be made instead of a flattening test. As at this time it was desired to take the views of the tube makers on some other points, this proposal was included in the reference.

The eight principal tube makers were therefore referred to in 1910, and they were also asked, among other things, whether they would propose any modification in the method of carrying out the specified flattening test.

The replies indicated that three of the firms had no objection to a tensile test if applied to 1 per cent. of the tubes, but all the remainder were strongly in favor of the flattening test being retained, but made various suggestions for making the test with lever presses or drop hammers instead of the hand hammer.

In the words of one firm, "it would be a great mistake to do away with the flattening test, which is an excellent proof of proper alloying." This firm further stated that the present specified impurities allowed, viz: 0.625 per cent., might with advantage be reduced by one-half.

From the above it appears that the majority of the firms were in favor of the flattening test, and the Admiralty saw no reason for abandoning it.

It may be asked what has been the practical result of these efforts on the part of the Admiralty and their tube-making collaborators?

My opinion is that the most beneficial action taken by the Admiralty during recent years is the practice adopted in 1904 by which a test piece was allowed for on each tube for the flattening test, a small piece being left on extra to the proper length, these pieces being sawn almost through, but not detached from the tube, and subsequently removed.

It is impossible to give the number of failures on service in the fleet in

detail, except in recent years. The earlier failures were very numerous. The following detailed figures are available as evidence that they have been reduced to more satisfactory proportions. In the year 1908, when the approximate number of condenser tubes at work at sea in the British fleet was 2,500,000, the number of failures of tubes occurring on service and reported were: Splitting, 21; corrosion and perforation, 69; total, 90, or at the rate of 1 in 28,000 per annum.

During the next five years there was a very considerable addition of new condenser tubes in the ships added to the fleet, owing to the great increase in horsepower of modern ships, also a considerable number of old tubes passed out of the service. In the year 1913 the number of condenser tubes at work in the British fleet was approximately 3,800,000, and the total failures of tubes reported during the two years 1912 and 1913 were: Splitting, 16; corrosion and perforation, 115; total, 131, or at the rate of 1 in 60,000 per annum—a very substantial reduction.

A further point is that of this proportion of 1 in 60,000, the majority are probably the older tubes which have been made under early specifications, and which have not had the present stringent regulations as regards treatment during manufacture and inspection applied to them. As these older ships and tubes fall out of service the proportion of tubes which fail may be expected to be less even than this small figure.

The ideal to work for is to reduce the failures to zero, but beyond a certain point further tests, restrictions and additional precautions, both in treatment when at work and during manufacture, becomes so expensive as to be not worth the cost involved, in which case we must be satisfied with the success already achieved.

In view of the number of failures of condenser tubes caused by corrosion and perforation by salt water where the circulating water passes through the tubes, close attention has been recently given to endeavors to eliminate the causes of these perforations. An incipient cause of some of the cases appears to be the existence of original flaws or breakages of the inside surface of the tube, and to endeavor to eliminate such tubes the Admiralty have recently required a rigid internal examination. The result of this internal sighting has been to indicate defects in the tubes, and such tubes on being cut up have, in a large number of cases, revealed imperfections such as to justify their rejection, and which, if these tubes had been fitted in place, would probably have caused early perforation.

I think I have shown that the Admiralty has endeavored to combine the observations and experience of the user and the experimental work of the laboratory with the commercial possibilities of the manufacturer, with the object, first, of securing reliability in the machinery of warships, reliability being of permanent importance in the time of war; and, secondly, of securing that reliability with the minimum of cost.

Similarly, in this Institute we hope to see the same coöperation and coördination of the work of the three main groups of members, in striving for the elucidation of all the problems connected with non-ferrous metals and their working and use. I have devoted some time to the condenser tube, as it has been the subject of special consideration by the Institute, but there is a large field for work for the members in other directions.

In order to obtain the best materials commercially possible to meet the requirements of design, there is a great deal to be ascertained as regards such matters as the obscure cases of corrosion of copper pipes, and of pipes of other non-ferrous alloys; the characteristics and peculiarities of alloys subjected under working conditions to stress under high and varying temperatures; the question of suitable alloys for propeller blades, especially with high speeds of revolution; and in many other directions.

There is therefore a large field for research work of obvious practical utility, and as among the various members of this Institute, differing as they do in their methods and thoughts, but working towards the same

ends, there exists a large body of scientific, experienced, and observant men, their united efforts should aid largely in the solution of such problems, which will be for the general good of the engineering and industrial world. I hope that our present meeting may bear fruit in this direction.—"The Engineer."

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## CORROSION OF IRON AND STEEL STRUCTURES AND THEIR PRESERVATION.

By Mr. F. Crosby-Jones.

At a meeting of the Institute of Marine Engineers held at Stratford recently, Mr. F. Crosby-Jones read a paper on the Corrosion of Iron and Steel Structures and their Preservation.

The author pointed out that some of the agencies which hasten the corrosion of iron and steel are: (1) Alternate exposure to action of atmosphere and water; (2) the deposition of water in depressions, cracks and crevices; (3) exposure to smoke, heated vapors and acid gases; (4) damp and vitiated atmosphere; (5) the accumulation of soot, dust, decaying matter, &c.; (6) dripping water, fresh or salt; (7) stray electric currents and electrolytic action; (8) several different classes of iron and steel incorporated in one structure; and (9) inequalities of temperature.

In some instances the agencies may be guarded against; but in many cases it is quite impossible to prevent them being present, therefore one must look to the improvement in the power of the metal to combat corrosion or to a suitable protective covering.

Painting is the method of protection against corrosion that has the most extensive use, owing to the fact that paint is easy of application, and as a product is cheap; further, it has the advantage of being readily renewable to structures where all other methods are impossible. Paints applied to iron and steel are engineering materials, and as such deserve more study and consideration by engineers; the "factor of safety" of iron and steel takes the effects of corrosion greatly into consideration; therefore, if more care be expended upon the surface the factor might be lowered somewhat in certain cases, provided that sufficient care is given to surface preservation. Paint is not a destroyer of rust, nor will it last forever, and will only protect iron or steel so long as it remains an adhesive and impervious coating. All paint undergoes alteration as it absorbs oxygen from the atmosphere. The pigment used may accelerate this absorption. With a good paint on application, the oxygen absorbed is 10 to 15 per cent. of the weight of the oil used in the constitution of that film. The properties of a good paint are, that it shall have no ill effects upon the metal itself, adhere firmly, be easy of application, be hard, tough and elastic, adapting itself to changes of temperature, and form a smooth and even surface impervious to air, water and gases. Paint should not remain soft underneath and harden on the surface, and should not be softened by subsequent coats of paint, nor should it flake off, powder or blister, but gradually wear away at its life comes to an end. The duration of the life of paint varies considerably. On one occasion the author had an engine-room bulkhead chipped and scraped for the purpose of repainting. The ship was built of iron and was 23 years old at the time. The coats of paints applied during service were readily removed, but the paint applied by her builders, on the engine-room side of the bulkhead, could not be readily removed, as it was so tough and adhesive, seeming to be ingrained to the metal. The iron on the engine-room side was in a splendid state of preservation, except where it had been rusted through from the alleyways and tops of the bunkers; certain repairs were being

executed at the time for this reason. The paint on the bunker side of the bulkhead was red lead ⅛ inch thick. This could be chipped off in large flakes, leaving the metal quite bare. The value of paint depends upon the nature, quality and preparation of the oils used; the character and proper selection of the pigment, and the care exercised in grinding and mixing, all of which mean expert knowledge, constant attention, good machinery and considerable expense; hence cheap paints are not to be relied upon. A curious feature that may often be noticed is, that a man will use paint for decorative purposes at three or four times the price he will pay for paint to protect iron or steelwork. A great deal depends upon the manner in which painting is done and the preparation of the work before commencement. It should be perfectly clean, free from grease, oil or moisture, and all scale and rust should be removed. The most efficient method of cleaning iron or steel is by pickling, but unless all trace of acid be removed after pickling it might be a dangerous means of inciting corrosion; it is, however, not applicable to large structures. Sand-blasting with dry, and not too sharp sand is the most efficient means of cleaning metal in structural work that is already erected. Failing the last method, all scale should be removed with scaling hammers that are not so sharp as to leave marks upon the solid metal, then steel scrapers and wire brushes used to remove all traces of rust, until the metal shows clean. All oil or grease should be removed by turpentine or benzine. Much painting is done upon metal when in a moist condition, and under such circumstances it is not to be wondered at that good results are not obtained. Wet and damp days should be avoided, the best results will be obtained when the metal is warmer than the surrounding atmosphere, thus avoiding the liability of condensation of moisture in the atmosphere upon the metal. Good stiff brushes should be used, and the paint well brushed into the metal. A coat of paint should be well brushed out, and should be as thin as possible with due regard given to covering power. One coat of paint should be thoroughly set, not skinned over, before another coat is applied over it, otherwise the paint will soon begin to crack. The usual long-handled brushes used in painting the hulls of ships are not at all good from a painting point of view, quite apart from the fact that they are the means of wasting much paint on the bottom of the dry dock, the reason being that the work is too far from the operator's eye for him to see if the surface is properly covered and the paint well worked into the surface.

Opening the discussion, Mr. Shanks said that the repainting of ships was generally done under adverse conditions. To begin with, there was the pressure of time; secondly, many engineers, with no knowledge of paints, applied the wrong ones indiscriminately over various parts of the ship; and thirdly, they often had to put up with inferior paint which was given to them to use. He thought more highly of bitumastic enamels than Mr. Crosby-Jones appeared to do. It was particularly effective for tanks under boilers.

Mr. E. W. Ross also spoke of the inferior paints given to engineers with which to do their best. He differed from the author of the paper when he said that pitting was not the same as corrosion. Mr. W. Walker said he had seen many wrought-iron boilers 40 years old in good condition. They were not built now, because the scantlings were necessarily heavier and became more brittle than in the case of steel. He could not agree with Mr. Crosby-Jones when he said that Portland cement was not very effective against corrosion. Tar on heated metal also reduced the danger.

Replying to the various questions, Mr. Crosby-Jones said that he considered that corrosion was a rust which spread evenly over the surface, whereas pitting was local. He admitted that if the metal were thoroughly

cleaned Portland cement was an effective anti-corrosive; but there was a danger of air and moisture getting between the cement and the metal.

The meeting adjourned until February 9, when the discussion was concluded and the author thanked for his contribution.—"Shipbuilding and Shipping Record."

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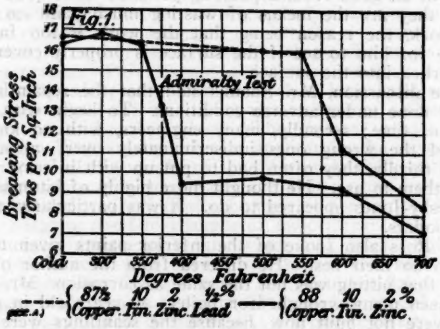
## BRONZE.*

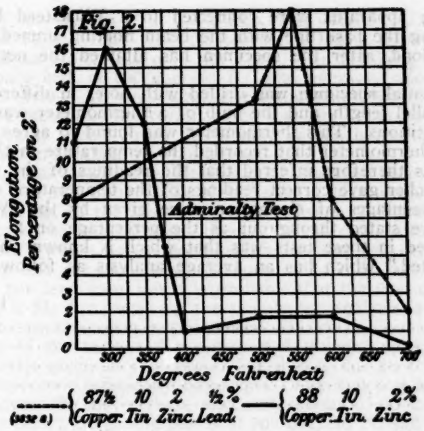
By John Dewrance.

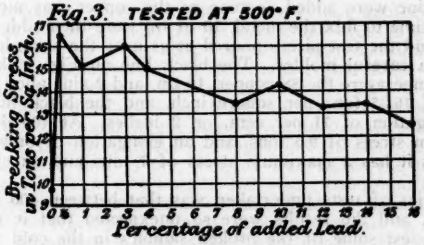
The dictionary meaning of the word "bronze" is a compound or alloy of from 2 to 20 parts of copper to 1 of tin, to which other metallic substances are sometimes added, especially zinc. In times past the gradually increasing additions of zinc and lead discredited the name of "bronze." In the manufacture of guns it was found that the best results were obtained by an alloy of 9 parts of copper and 1 part of tin. This became the standard material for the manufacture of guns for many years. To distinguish this alloy from the inferior mixtures that had previously been supplied under the name of bronze, the description "gun-metal" was introduced. As time went on the new name, "gun-metal," was no more respected that the old one of "bronze," and at the present time any alloy that does not come under the description of pot metal or brass is called "gun-metal." As guns are now universally made of steel, it seems desirable to return to the dictionary description, and to call all alloys mainly composed of copper and tin "bronze." To the previously-mentioned alloy of 90 per cent. copper and 10 per cent. tin, it is very largely the practice to add 2 per cent. of zinc, making 88 per cent. copper, 10 per cent. tin, and 2 per cent. zinc. When tested at atmospheric temperature this alloy gives very excellent results. As a great deal of bronze is used at the temperature of high-pressure steam, it becomes important to investigate its behavior at temperatures that correspond.

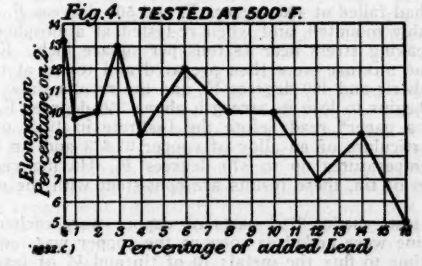



The tests, the results of which are given in Figs. 1 to 4, were conducted for the author by Mr. R. H. Harry Stanger. The heating apparatus was an airtight tube boiler, heated by gas from a ring burner. The specimen was held in screwed jaws in the center, both end jaws being insulated, and the boiler was also coated with asbestos. The testing machine being a 50-ton vertical Buckton, the weight of the apparatus is carried on the bottom

*Paper read before the Institute of Metals, March 18, 1914.




headstock; the top end of the specimen was held in a plunger which enters the top of the heating apparatus through a broad guide with a sliding fit. Both ends of the apparatus are held in the headstocks by means of spherical holders, thus allowing the whole to find its true vertical axis.

As the specimen is entirely enclosed during the test, some outside means have to be adopted for ascertaining the yield point, if any; the two ends

of the heating apparatus were connected to a Wicksteed hydrographic recorder, taking the base line with the beam floating immediately before applying the load, after the specimen has attained the necessary temperature.

An experimental specimen was drilled with holes in different positions along the parallel length, and the bulb of a thermometer was inserted in the various positions. This thermometer was found to agree very closely with another thermometer that recorded the temperature in the air chamber, and it was therefore inferred that the readings of the thermometer in the air chamber gave correct readings of the temperature of the specimen. The percentages of elongation were given by the Wicksteed recorder, and are stated throughout as the percentage on 2 inches. The copper employed in these tests was that which is known on the market as "best selected," which has an average analysis as follows:

| | Per cent. |
|---|---|
| Copper | 99.55 |
| Nickel | 0.01 |
| Arsenic | 0.026 |
| Lead | 0.08 |
| Bismuth | 0.004 |

The tin and zinc were the best commercial quality.

In the first tests made 88 parts of copper were melted in a new crucible, two of zinc were added as soon as the copper was melted and allowed a short time to flux the metal, 10 of tin were then added, the whole mass stirred, and the test pieces poured at as near the same heat as could be judged by a careful molder. The black line on Fig. 1 shows that at atmospheric temperature the 88 copper, 10 tin, and 2 zinc alloy has a maximum stress of 16.35 tons per square inch, and the black line in Fig. 2 shows an elongation of 11 per cent. on 2 inches. At 400 degrees F. it has a maximum stress of 9.5 tons, and an elongation of only 1 per cent. At 700 degrees it has a maximum stress of 7 tons and an elongation of 0.25 per cent.

The first series of tests undertaken was that between 400 degrees and 700 degrees F., and the results were so unexpected that it was thought advisable to re-test some of the broken samples in the cold state, to ascertain if the fault was in the casting of the test pieces. For this purpose samples which had failed at 400 degrees F. and 500 degrees F. were turned down and suitably mounted, and when re-tested at atmospheric temperature gave a breaking stress near 18 tons per square inch. Further samples in this same mixture were then prepared and tested at temperatures between atmospheric and 400 degrees F., and the results show very clearly that the metal begins to lose its strength above 350 degrees F. Professor Huntington, in a paper* read before the Institute in 1912, gives, among other alloys, particulars of an alloy of copper 97.673 and tin 2.408, tested cold, and at temperatures up to 870 degrees F. Having regard to the small proportion of tin, these results are consistent with the results given above.

In the next tests made, 87½ parts of copper were melted in a new crucible, 2 of zinc were added as soon as the copper was melted and allowed a short time to flux the metal; 10 of tin and ½ of lead were then added together, and the whole mass stirred.. It will be observed that the only difference between this and the previous experiments is the addition of ½ per cent. of lead at the expense of the copper. The dotted line on Fig. 1 shows that at the atmospheric temperature the maximum stress is 16.5 tons per square inch, and the dotted line in Fig. 2 the elongation, 8

*Journal of the Institute of Metals, 1912, No. 2, vol. viii., page 131. ENGINEERING, September 27, 1912.

per cent. At 550 degrees F. it has a maximum stress of 15.8 tons and an elongation of 18 per cent. At 700 degrees it has a maximum stress of 8.25 tons and an elongation of 2 per cent. The breaking stress of 11.25 tons per square inch at 600 degrees F. is an average. No actual sample broke at this stress. It was found that some samples tested at this temperature gave results in the region of 16 tons per square inch, and others 7 tons per square inch. From this it may be concluded that 600 degrees is the critical temperature of this alloy.

In a paper read before the International Association for Testing Materials at the Seventh Congress in New York, in 1912, by J. M. Bregowsky and L. W. Spring, of the laboratory of the Crane Company, Chicago, the results are given of tests, among others, of a material called "U. S. Navy gun-bronze G," which has a composition of 87.6 per cent. copper, 10.4 per cent. tin, 1.31 per cent. zinc, 0.39 per cent. lead; it is also stated to contain 0.11 per cent. iron, but it is probable that the iron content given is due to using a file for preparing the sample for analysis, as such an alloy ought not to contain such a proportion of iron.

The chart of the test of this metal at first sight appears inconsistent with the results given in this paper, but this is due to the fact that the tests made were not sufficiently numerous. The first test appears to be at about 80 degrees F., and gives a maximum stress of 15 tons and elongation of 9 per cent.; the second test is at 300 degrees F., and gives a slightly increased maximum stress of 16½ tons and elongation of 9.5 per cent. The next test is at 450 degrees F., and gives a maximum stress of 14.75 tons and elongation of 8 per cent. There is not another test until 600 degrees F., at which temperature the maximum stress is given as 10 tons and the elongation as 4 per cent. As previously mentioned, 600 degrees F. is the critical point, and it is unfortunate that there is such a wide gap of temperature between this test and the previous ones, as otherwise the results obtained would have been more consistent with the results given in this paper, and might have given information as to the slight difference due to that particular composition of alloy tested.

If it is accepted that ½ per cent. of lead raises the maximum stress at 500 degrees F. from 9.75 tons to 16.5 tons, that proportion of lead becomes an essential ingredient in bronze, that is subjected to temperatures above 350 degrees F. It also becomes necessary to inquire whether any further advantage can be gained by adding a larger proportion of lead. Figs. 3 and 4 show that this is not the case; but it is surprising that with so large a proportion as 16 per cent. of lead the maximum stress at 500 degrees F. is 12½ tons, as against 9.5 tons without any lead. In making these experiments the lead was added at the expense of the copper; so the last result would be 72 per cent. copper, 10 per cent. tin, 2 per cent. zinc, and 16 per cent. lead. In making the foregoing experiments the question arose as to whether any of the results were due to the absorption of oxygen from the atmosphere by the alloy when in a molten condition. It is difficult to determine the content of oxygen in bronze, so the following experiments were conducted with best selected copper:

*Experiment No. I.*—A new 100-pound plumbago crucible was taken. As soon as the copper was sufficiently hot an ingot was cast, and the rest of the metal was returned to the fire for an hour, when a second ingot was cast, and the same thing repeated for the third ingot. There were thus three ingots cast from the same crucible at intervals of one hour. On analysis the oxygen contents of the three were as follows:

| | Oxygen. Per cent. | Cuprous Oxide. Per cent. |
|---|---|---|
| No. 1 | 0.032 | 0.285 |
| No. 2 | 0.272 | 2.499 |
| No. 3 | 0.404 | 3.605 |

*Experiment No. II.*—Two new plumbago crucibles containing 100 pounds of B.S. copper were taken, and ¼ pound of 10 per cent. phosphor-copper was added to one, and 0.6 ounce aluminum to the other, as deoxidizers. The copper was covered with glass, so that when molten no air could come in contact with it; and three ingots were cast at intervals of one hour, as in the former experiment, the glass being left on during the casting.

| *Oxygen Contents on Analysis.* | | Oxygen. Per cent. |
|---|---|---|
| Aluminum as deoxidizer..... | No. 1 | 0.084 |
| | No. 2 | 0.080 |
| | No. 3 | 0.092 |
| Phosphorus as deoxidizer..... | No. 1 | 0.036 |
| | No. 2 | 0.036 |
| | No. 3 | 0.036 |

Both sets of results show that if the metal is kept from contact with the air when molten no oxygen is picked up; and if this precaution is not taken each melting will result in an increase in the content of oxygen, and this conclusion was accepted and acted upon when making the following experiments. The products were not analyzed for oxygen on account of the previously mentioned difficulty of such analysis.

*Experiment No. III.*—One new plumbago crucible, containing 100 pounds of 87½ per cent. copper, 10 per cent. tin, 2 per cent. zinc, and ½ per cent. lead, was covered with glass to exclude the air, and melted; another exactly similar charge was allowed ample opportunity to pick up oxygen by being repeatedly skimmed and left standing in the air after being taken from the fire. Test pieces, and a heavy flange casting with a light and thin nipple on each side, were cast from each crucible. The flange castings were broken and examined very carefully, and both appeared as sound as it is possible to get a casting. The test pieces gave the following results when tested cold and under heat:

| | Temperature. | Maximum Load. Tons p. sq. in. | Extension on 2 inches. per cent. |
|---|---|---|---|
| First charge covered with glass to exclude oxygen ..... | Cold | 15.17 | 9.0 |
| | At 500° F. | 13.73 | 10.5 |
| Second charge uncovered to absorb oxygen ..... | Cold | 16.55 | 12.5 |
| | At 500° F. | 15.15 | 14.0 |

*Experiment No. IV.*—To complete the above results Experiment No. IV was made, which consisted in casting a similar flange and test pieces in "twice run" metal. The flange casting again proved sound, and the test pieces gave the following results:

| | Temperature. | Maximum Load Tons p. sq. in. | Extension on 2 inches. per cent. |
|---|---|---|---|
| Metal twice run without any precaution to prevent absorption of oxygen ..... | Cold | 15.19 | 8 |
| | At 500° F. | 16.15 | 14 |

The foregoing experiments point to the conclusion that in an alloy of 87½ per cent. copper, 10 per cent. tin, 2 per cent. zinc, and ½ per cent. lead, no benefit results from the deoxidation of the metal, or in taking special precautions to prevent the metal absorbing oxygen.

There is another feature of considerable interest in these alloys. When the alloy that is free from lead is turned in the lathe, the turnings have a considerable curl upon them. The addition of ½ per cent. of lead ma-

terially reduces the length of the turnings. It would naturally be supposed that this feature is due to the fact that without the lead the alloy has an elongation of 11 per cent. on 2 inches, and with the addition of the lead the elongation is reduced to 8 per cent., and that the turnings break off more readily for this reason. The same test pieces from which the turnings were made when cold were then heated and turned at 550 degrees F., and it was seen that the alloy without the lead still presented the same curly appearance, and the turnings from the alloy containing the ½ per cent. of lead were very little longer than the turnings produced at the cold.

The elongation of the alloy containing ½ per cent. of lead, as will be seen by the chart, is at 550 degrees F., 18 per cent. on 2 inches; whereas the elongation of the alloy without the lead falls at this temperature below 2 per cent. As lead has such a marked effect on bronze, enabling it to be used without loss of strength up to 550 degrees F., it seems reasonable to expect that some other metal might be added that would enable the bronze to withstand even higher temperatures. With this object in view, 87.25 copper was melted in a new crucible, 0.25 of silver was added when the copper was melted, and the whole mass stirred. Ten per cent. of tin and 2 per cent. of zinc and ½ per cent.. of lead were then added and again stirred. Two test pieces were prepared and tested at 700 degrees F. One test piece gave a maximum load of 8.11 tons, the other 8.75 tons. The extension in neither case exceeded 0.5 per cent. on 2 inches. The maximum load is practically the same as without the silver, and the extension not so good. Nickel is not very promising, as in small proportions it seems invariably to liberate some occluded gas on cooling, and produces very porous castings. Aluminum is objectionable, because in even small proportions it seems to add materially to the amount of the contraction of the casting on cooling. Iron is strongly objected to in fine bronzes, as it combines with the tin and separates out into hard masses in the casting. The subject seems to call for further research, as it is possible that small proportions of some of the rarer metals may have a beneficial effect. With the present use of superheated steam it is very desirable that a bronze should, if possible, be produced that can be used with safety at 700 degrees F.—"Engineering."

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## WRITING SPECIFICATIONS FOR NON-FERROUS METALLIC PRODUCTS.

The practice of purchasing materials to specifications now so generally employed by large consumers has called attention to the difficulties surrounding the writing of specifications for wrought non-ferrous metallic products, such as brass and copper tubes, sheet, rod, wire, etc.

The purpose of a specification is to accurately define the exact characteristics, physical and chemical, which it is requisite that the material should possess, the limits of departure from these characteristics which can be tolerated, and the tests to be employed in determining them.

It is natural in drawing such specifications that the precedents established by the specifications for structural steel in similar forms should be followed, but a careful consideration of the problem indicates that there are such very important differences between the two classes of material that the analogy does not hold. In connection with the use of structural steel, the ability to withstand stress without deformation is the quality of prime importance. Extensive, minute and comprehensive investigations have determined the chemical composition best suited for each of various specific requirements, the range of variation needed to meet the limitations

of mill practice, and the physical properties as measured by standard tests corresponding to such compositions.

Considerations other than ability to withstand stress, however, generally determine the value of wrought non-ferrous metals and alloys. In the case of copper, its high degree of electrical conductivity is for obvious uses, the factor of prime importance. In other employments, its resistance to the corrosive action of various agencies is controlling, while strength or toughness may be valuable contributing qualities. With the brasses and bronzes the requirements of a wide range of usage determine the qualities of dominating importance. Seamless tubes for condensers employing sea water as a cooling agent demand primarily ability to withstand the corrosive and disintegrating action of that liquid, while other considerations are largely subordinate. Similarly, sheet brass in addition to its non-corrodible qualities derives its value from its ability to withstand without rupture the distortion and strain incident to its fabrication into articles and wares of various sorts.

It will be seen therefore from the above considerations that the framing of adequate and rational specifications for these materials is a problem requiring for its solution a vast amount of knowledge of the requirements of various specific uses, the physical and chemical properties which will satisfy those uses in the highest degree, together with conclusive tests by which these properties may be measured. In addition, the limitations imposed by markets or by the requirements of mill methods, practice and equipment must be quantitatively understood. No paralleling of specifications of material thought to be analagous can do other than produce inadequate and confusing results.—"The Brass World."

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## FRICTION IN BEARINGS.

As was pointed out by the author of the paper entitled "High-Speed Bearings," recently presented at a meeting of the North-East Coast Institution of Engineers and Shipbuilders, it is known that so far as friction is concerned it is not advisable to use a large shaft in a long bearing, but experience shows that for high speed and heavy loads long bearings are necessary, and they have also to be large to avoid vibration and critical speeds and also the risk of fracture due to vibration. To avoid this latter danger the bending stresses have to be kept to a figure which would be considered very low in ordinary practice, as there have been several cases of fracture of shaft ends. If the pressure is too high, or the oil too thin, the bearing may run all right so long as there is no vibration, but if vibration occurs the oil film may be driven out and seizing takes place. This effect is reduced in very high-speed shafts, as for instance, those running above 2,000 r.p.m., by the use of the well-known Parsons' flexible bearings where there are two or more tubes outside the bearing bush with a small clearance between them in which the viscosity of the oil tends to damp out vibrations. In turbine work a rule much employed for deciding the size of the bearings has been that the product of the speed by the pressure should not exceed a certain figure, but it is doubtful whether this rule is of much value, as it really assumes greasy friction and not perfect lubrication; in land work, a product of 5,600 has rarely been exceeded, or 75 pounds per square inch at 75 feet per second. At lower speeds higher pressures are used satisfactorily, as, for example, at about 35 feet per second, a pressure of about 130 to 150 pounds per square inch is permissible.

## HIGH TEMPERATURES IN CAST-IRON CYLINDERS.

In the construction of internal-combustion engines the thickness and quality of the material used for cylinders and pistons and the effects produced by very high temperatures are matters which have to be very seriously considered. It is quite conceivable that a thick cylinder opposed to high temperature as well as to a high pressure inside it will be actually more stressed than a thinner one. The same reasoning applies to the piston crown and the bottom of the cylinder covers. As was pointed out by Mr. J. T. Milton in his paper, the "Present Position of Diesel Engines for Marine Purposes," the metal in these parts is put into a state of compression by the high state of the temperature, the sides being subject to a similar stress due to the pressure. These considerations show the great importance of keeping the thickness of the metal at parts exposed to great heat as thin as possible consistent with their possessing sufficient strength. It is known that cast iron appreciably changes its form and dimensions through continuous high temperatures, and so causes troubles similar to those which have occasionally been experienced in large steam engines. It is probably the fear of this action under severe heat conditions which led to the use of cast steel for cylinder covers. Most Diesel-engine manufacturers have reverted to the use of cast iron for covers, and some of them give the metal a severe heat treatment before the completion of the machining operations. It is quite possible, however, that cast steel will be again used for these parts.

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# BOILERS.

## TREATMENT OF BOILER FEED WATER.

Report by Professor Blacher, of Riga, on Harmful Ingredients Contained in or Added to Boiler Feed Water, and on the Influence of the Working Pressure on the Action of the Said Ingredients.—The presence of carbon dioxide in the boiler feed water cannot be considered objectionable as a rule, but where the process of corrosion of the boiler sheets has already started it may accelerate it. It can, however, be easily removed by caustic soda. Air must be considered as a harmful agent, since the protective action of oxygen in boiler practice has been neither observed nor proved. The air in the presence of chlorides and under certain conditions causes dangerous spot corrosion, and, since it is due to the action of the oxygen of the air mechanically carried into the boiler with the feed water, it is advisable to take care to admit as little as possible. It is very desirable to admit the feed water either into the steam space or closely below the water level, so as to permit the air to escape into the steam. If there is a large amount of chlorides present in the water it appears advisable to reject such water entirely, but, if not, it is well to observe the inside of the boiler very carefully, and as soon as signs of corrosion appear, to install zinc plates in metallic connection with the boiler shell, taking care, however, to see that as little as possible of air is allowed to enter, since otherwise the oxygen will in a very short time destroy the zinc. As regards the organic constituents, Reischle, of Munich, discusses particularly the action of various tar, asphalt and graphite paints, and mentions the Brun process, widely used in Switzerland, which consists in the addition to the boiler water of some linseed oil. On the whole, he comes to the following conclusions with respect to the use of means for preventing scale: They prevent more or less formation of scale, but cannot be considered as a certain method of protection,

with the exception, perhaps, of the above-mentioned Brun process; they produce solution of old scale only occasionally, but usually make it brittle so that it either breaks off of itself or can be more easily removed. Sometimes their application may lead to overheating and injurious results to the boiler plates, especially the firebox plates, and, finally, they can produce clogging of the water-level glass and blow-off cocks.—"Journal American Society of Mechanical Engineers."

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## WORKING UP A BOILER TEST.

By F. R. Low.

SYNOPSIS.—*Explaining the computation of the results of a capacity test. Equivalent evaporation from and at 212 degrees. Factors of evaporation. Correcting for moisture, superheat and leakage. Percentage above or below rating.*

A boiler test may be made simply to find the capacity of a boiler, or to determine the efficiency, or it may be extended so as to furnish the information needed for an analysis of the performance and to show why it fails to produce desired results.

### TEST FOR CAPACITY.

One would naturally say that the capacity of a boiler is measured by the amount of steam which it will make in a given time. This is true—with modifications. The same boiler will make more steam per hour with hot feed water than with cold, it will make more "wet steam" than it will dry or superheated steam, and it will make more (though not so much more, as many think) of low-pressure than of high-pressure steam. In order, then, to determine the real capacity of a boiler it is necessary to know the: *a* duration of the test; *b* amount of water fed to the boiler corrected for difference in level at the beginning and the end of test; *c* temperature of feed water; *d* pressure of steam; *e* quality of steam.

The item *a* is simply the time during which the boiler was run at the capacity which it is sought to measure.

The item *b* is the amount of water which the boiler got rid of, but it may not all have been converted into steam. A part may have been carried over as unevaporated water. To turn out steam which was exactly "dry saturated," which had neither moisture nor superheat would be a feat for a laboratory, so that if the test is to be accurate the quality of the steam, item *e*, must be known. Some of the water may have leaked out, but we will, for simplicity, suppose the boiler and connections to be tight below the water line.

Suppose a capacity test has yielded the following results:

| | |
|---|---|
| Duration of test, hours | 10 |
| Amount of water fed to boilers, corrected for difference in levels, pounds | 137,642 |
| Temperature of feed water, degrees | 196 |
| Pressure of steam (gage), pounds | 122.6 |
| Quality of steam, per cent | 96.5 |

Of the 137,642 pounds of water fed to the boilers only 96.5 per cent., or

$$137{,}642 \times 0.965 = 132{,}824.53 \text{ pounds}$$

was evaporated; and on an average one-tenth of this, or 11,282.453 pounds per hour.

The steam pressure given, 122.6 pounds, is the average for the 10 hours of the test; that is, the way that the fractional pounds come in; not that the gage has been read to tenths of a pound. It is the "gage pressure" or the pressure above that of the atmosphere. The pressures used in the steam tables are absolute pressures, *i.e.*, gage pressures plus the pressure of the atmosphere. At the sea level the pressure of the atmosphere varies around 14.7 pounds; at an elevation it is less. If one has the reading of the barometer during the test he reduces it to pounds per square inch by multiplying the barometric reading in inches of mercury by 0.49 and adds the pressure so found to the gage pressure. In the absence of the barometric reading add the usual pressure of the atmosphere at the place at which the test is made. Suppose the test to be made at the sea level we add

122.6 gage pressure.
14.7 atmospheric pressure.

137.3 pounds absolute pressure.

It will make little difference in the result if we call the average absolute pressure 137 pounds, but the values of the properties of steam are given in the table in even degrees as well as in even pounds. It may be possible that the pressure corresponding with some even degree will be 137.3 or nearer to it than 137, the nearest value in the pressure table. The temperature corresponding with 137 pounds is 351.4. Look at this part of the temperature table and find that 351 degrees corresponds with 136.3 pounds and 35[illegible]egrees with 138.1 pounds. There is nothing between, and neither is near the given value 137.3 than the line for 137 pounds on the pressure table.

The table* shows in the "total heat" ($H$) column that it takes 1191.8 B.t.u. to make a pound of dry-saturated steam of 137 pounds pressure out of a pound of water at 32 degrees. Turn to 196 degrees the feed-water temperature in the temperature table and find in the "heat of the liquid" ($h$) column that it takes 163.92 B.t.u. to raise a pound of water from 32 to 196 degrees. This much is already in the feed supplied at 196 degrees. The boiler has had to put in the difference,

1191.80 B.t.u. $= H$ at 137 pounds.
163.92 B.t.u. $= h$ at 196 degrees.

1027.88 B.t.u. required per pound evaporated from 196 at 137 pounds.

Water boils at the usual atmospheric pressure, 14.7 pounds absolute, at 212 degrees. Look at the temperature table, and in the line for 212, seventh column marked "latent heat of evaporation" ($L$), find the value 970.4. This means that after you get the pound of water up to the boiling point, 212 degrees, you have to put 970.4 B.t.u. into it to boil it away. If you had a pound of water in a flask at 32 degrees under 14.7 pounds per square inch absolute pressure and applied heat to it, it would take 180 B.t.u. to raise it to 212 degrees, when it would commence to boil, and 970.4 B.t.u. or nearly five and one-half times as much heat would be required to boil it away. If the flask absorbed heat at the same rate all the time it would take over five times as long to make the water into steam after it began to boil as to heat it to the boiling point. This evaporation is taking place "from and at 212 degrees," *i.e.*, from feed water of 212 degrees temperature into steam of 212 degrees temperature and 14.7 pounds pressure, and this evaporation from and at 212 degrees requiring

*The values given are from the Marks-Davis tables.

970.4 B.t.u. per pound is the standard to which evaporations from other temperatures of feed and at other pressure are referred for comparison.

We have seen above that it takes 1027.88 B.t.u. to make a pound of steam under the conditions of the supposed test. Then each pound of steam made under these conditions takes

$$1027.88 \div 970.4 = 1.0592$$

times as much heat as it does to make a pound of steam "from and at 212 degrees." This figure 1.0592 is called the "factor of evaporation" for the given condition. Tables of them are to be found in most engineers' reference books. The factors are found by dividing the difference between the total heat $H$ of the steam at the given pressure and the heat of the liquid $h$ at the feed-water temperature, by 970.4.

$$\text{factor of evaporation} = \frac{H - h}{970.4}.$$

We have found above that the boiler under test evaporated 13,282.453 pounds of water per hour from 196 degrees and at 122.6 pounds. This would be equivalent to

$$13{,}282.453 \times 1.0592 = 14{,}068.774 \text{ pounds}$$

from and at 212 degrees.

The evaporation of 34.5 pounds per hour from and at 212 degrees is the requirement per boiler horsepower. The horsepower developed by the boiler on test was therefore

$$\frac{14{,}068.744}{34.5} = 407.79 \text{ boiler horsepower.}$$

This boiler horsepower corresponds to the absorption of 33,478.8 B.t.u. per hour. The horsepower could be found then by multiplying the evaporation in pounds per hour by the British thermal units required per pound evaporated, and dividing the number of British thermal units absorbed per hour thus found by 33,478.8 thus:

$$\frac{13{,}282.453 \times 1027.88}{33{,}478.8} = 407.8 \textit{ boiler horsepower.}$$

The fundamental basis for the horsepower is that adopted by the Committee on Boiler Tests of the Centennial Exposition of 1876, 30 pounds from 100 degrees into steam of 70-pounds gage which requires 1115.33 B.t.u. per pound; and the evaporation of 30 pounds under these conditions requires

$$1115.33 \times 30 = 33{,}459.9$$

or, dropping the fraction, 33,460 B.t.u. per hour, which corresponds to

$$\frac{33{,}459.9}{970.4} = 34.48 \text{ pounds}$$

from and at 212 degrees. The Committee of Boiler Tests of the American Society of Mechanical Engineers recommended that this shall be considered equal to 34.5 pounds from and at 212.

To determine the percentage of the rated capacity divide the horsepower as determined by the test by the rated horsepower and multiply by 100. Thus if the boiler in the above test were rated at 350 H.P. it would have been running at

$$\frac{407.79 \times 100}{350} = 116.51 \text{ per cent.}$$

of its rated capacity or 16.51 per cent. above its rating.

If it were rated at 500 H.P. it would have been running at

$$\frac{407.79 \times 100}{500} = 81.56 \text{ per cent.}$$

of its rated capacity, or

$$100 - 81.56 = 18.44 \text{ per cent.}$$

below its rating.

WITH SUPERHEATED STEAM.

If the steam were superheated instead of moist it would be necessary only to use the value for the total heat of the superheated steam (which may be found in the Marks-Davis table for superheated steam) in the same process as above, except, of course, that there is no correction for quality. Suppose, for example, the conditions and results were the same as before, except that the steam instead of having the quality 96.5 per cent., *i.e.*, carrying 3.5 per cent. moisture, were superheated 100 degrees.

The table shows that to make a pound of steam of 137 pounds' pressure, superheated 100 degrees from feed water of 32 would take 1247.4 B.t.u. The feed water at 196 degrees has already in it 163.92 B.t.u. above 32. The heat required per pound is therefore

$$1247.4 - 163.92 = 1083.48 \text{ B.t.u.}$$

In this case all the water fed to the boilers

$$137{,}642 \div 10 = 13{,}764.2 \text{ pounds per hour}$$

was evaporated. The boiler, therefore, absorbed

$$13{,}764.2 \times 1083.48 = 14{,}913{,}235.416 \text{ B.t.u. per hour}$$

which is equivalent to

$$\frac{14{,}913{,}235.416}{33{,}460} = 445.7 \text{ boiler horsepower.}$$

CORRECTING FOR LEAKAGE.

Suppose that during the test the boiler or some of its connections below the water line, as the blowoff valve or pipe, commence to leak. It does not make any difference if the leak is where it will blow steam. If the boiler has made the steam, that is all we want. We do not care, so far as the purpose of the test is concerned, whether the steam blows out of the safety valve, or out of a leaky joint or goes to a condenser directly or through an engine. But if some of the water which has been credited to the boiler is got rid of by leakage instead of by evaporation, less heat has been absorbed and a correction must be made for it.

To make this correction the amount of leakage and the average temperature of the leaking water should be known. If the temperature is above 212, some of the water will turn into steam as soon as it gets outside the boiler, and it may be necessary to measure the quantity by measuring the drop of the water line in a given time, when no steam is being taken from the boiler and the pressure is constant at that of the test, or the rate of feed necessary to keep up with the leak. It may be hard to determine the temperature of the escaping water, which may not be that due to the pressure of the steam. But this article is upon working up a boiler test, not upon making one. Assuming a test with results as given above, 100 degrees superheat, 880 pounds' leakage; temperature of leaking water, 340 degrees; what is the horsepower?

We must subtract from the water credited to the boiler this 880 pounds which it did not evaporate.

$$137{,}642 - 880 = 136{,}762 \text{ pounds}$$

evaporated in 10 hours, or 13,676.2 pounds per hour. We found above that it required 1083.48 B.t.u. per pound, so that the water evaporated represents.

$$13{,}676.2 \times 1083.48 = 14{,}817{,}889.176 \text{ B.t.u.}$$

But in addition the boiler absorbed the heat, which was used to heat the water which leaked

1 pound at 340 degrees carries 311.00 B.t.u. above 32.
1 pound at 196 degrees carries 163.92 B.t.u. above 32.

147.08 B.t.u. added.

The 88 pounds leaked per hour would then have taken out of the boiler more than they brought in

$$147.08 \times 88 = 12{,}943.04 \text{ B.t.u.}$$

The boiler has absorbed the heat, and when the leak is stopped it will be useful for making steam. If we want to know the capacity of the boiler when it is not leaking we must credit it with this heat

14,817,889.176 B.t.u. used in evaporation.
12,943.04 B.t.u. lost by leaking.

33,478.8)14,804,946.136 B.t.u. absorbed per hour boiler H.P.

442.2

In the case of the boiler making moist steam, some of the heat which is absorbed by the boiler goes off in the entrained water which is heated from the feed to the steam temperature. If the steam is to be used for driving a turbine or engine this moisture is worse than useless and the boiler should get no credit for the heat which it carries off. If the steam is used for heating or processes this heat may be as useful as any other, and will be credited to the boiler or not, according to what you are trying to determine by the test.

When we come to making up a heat balance and finding out what has become of all the heat in the fuel this item will have to be reckoned with, but that is another story.—"Power."

## WASTE-HEAT BOILER.

On the evening of March 6, the Cleveland Chapter, N. A. S. E., was entertained at the works of the Bruce-Macbeth Engine Co., by J. B. Meriam, in a lecture on a new exhaust boiler which has recently been developed by that company. The photograph was taken during the lecture and shows a 150-H.P. four-cylinder gas engine equipped with an exhaust boiler. In the ordinary gas engine about 30 per cent. of the heat in the gas is wasted to the jacket water and about 35 per cent. in the exhaust cases. By circulating this water very rapidly through the jackets in connection with the boiler which utilizes heat from the exhaust gases a significant part of the waste heat is recovered in the form of low-pressure steam.

It is understood that the Case School of Applied Science is to conduct extensive tests on the boiler.—"Power."

## THE USE OF SUPERHEATERS AND SUPERHEATED STEAM IN MERCANTILE STEAMERS.

By Harry Gray.

Paper read before the Institution of Naval Architects April 2, 1914.—This paper is reprinted in "Engineering," April 10, 1914, and comment thereon also appears in the "Engineer," April 3, 1914.

This paper shows the actual economy secured by superheat on several merchant steamers and gives an interesting account of experiences encountered in using superheat in marine work.

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## OIL-FUEL EVAPORATIVE RESULTS.

Extract from Mr. Jas. S. Gander's article read before the Institute of Marine Engineers.

So far as the marine engineer is concerned the comparisons of the various consumptions per pound of oil used per indicated horsepower, do not satisfy. One vessel may be fitted with the latest feed-heating and superheating appliances, etc., while the others may be of an older type, possessing none of these advantages. If we consider, however, the amount of oil burnt per I.H.P. per hour, compared with the amount of coal burnt for the same power, the other conditions cancel out from both sides of the equation, and we are enabled to ascertain whether or no the oil fuel is being economically burnt. An average result gives the ratio of coal burnt to the weight of oil burnt for the same power as 3.2. This means a saving by weight of 33 per cent. and shows oil fuel to be 50 per cent. more effective. This does not agree with the respective calorific values. Taking average British coal with a value of 14,000 and average fuel with 19,000 B.t.u. the efficiency advantage should be about 36 per cent. The added efficiency which is obtained in practice is due to the more complete combustion of oil fuel. From the details of a test taken at the works of Smith's Dock Co., we gather some valuable information relating to temperatures which are not easily obtainable at sea. Dimensions of boiler 8 feet 9 inches by 8 feet 6 inches with two plain furnaces, 31 inches external diameter, 6 feet 1 inch long, total heating surface 470 square feet. Meyer Smith system used.

| | |
|---|---|
| Temperature of air, degrees F. | 54 |
| in passages round air conveyer in furnace fronts through which the air passes to furnaces, degrees F. | 870 |
| of furnace, degrees F. | 2,375 |
| of combustion chambers, degrees F. | 1,280 |
| of smoke box, degrees F. | 580 |
| of feed water, degrees F. | 120 |
| of oil in heaters, degrees F. | 220 |
| Pressure of oil to burners, pounds | 35 |
| steam on boilers, pounds | 80 |
| Consumption of oil per hour, pounds | 164.35 |
| Total water evaporated per hour, pounds | 2,592.51 |
| Water evaporated from and at 212 degrees F. per pound of oil used, pounds | 15.7 |
| Brand of oil used | 2 parts Texas, 1 part Borneo |
| Specific gravity | 944 |
| Flash point, degrees F. | 246 |
| Calorific value (not stated in table, but would be about) B.t.u. | 19,105 |

## ENGINES.

### UNITED STATES NAVAL COLLIER *JUPITER*.

**The following letter will no doubt be of interest to readers in this country. The claims made come as news to those in America. The *Jupiter's* machinery was developed in the United States independently of Mr. Durtnall's patent. Though of course the idea of bridge control and use of an electric motor for ships' propulsion have been in evidence for many years.**

76, Gladsmuir Road,

Whitehall Park, Highgate, N.
November 22.

To the Editor of the Shipbuilding and Shipping Record.

My attention has been drawn to a letter which appeared in your issue of November 20 last, from The British Thomson-Houston Company, Ltd., of London and Rugby, in which they state that your reference to my invention which appeared in your issue of the 6th inst. was not true. And, further, go on to state and give notice that they own and control the patents in this country of the General Electric Company of Schenectady, United States of America, under which the turbines and the electrical machinery installed in the *Jupiter* were made. Their statement is perfectly true; the above machinery is of the very best that could be constructed at the time, and is, indeed, as covered by many interesting patents, owned and controlled by the General Electric Company of America, and for whom the British Thomson-Houston Company, Ltd., are the British representatives.

But your statement that I was the inventor of the system employed on the *Jupiter* is correct. It was as chief engineer of the Cannon Street Hotel, London, that, in 1893 to 1895, I made a series of experiments, but I was not at the time successful in getting the idea taken up. Eventually I took out protection in April, 1903 (British No. 8137), and further improvements were brought out by me in August, 1905, and in this the system employed on the *Jupiter* was embodied. It was pointed out that great advantages would be found in the application of the system (since called the "Paragon" system) to the propulsion of ships. The last paragraph of the patent specification reads as follows:

"It is possible to control the ship from one or more positions, such as the captain's bridge, and also, by winding the electric motors with two or more magnetic poles, more than the generator, means are thus provided so that the electric generator can be run at very high speed, and the electric motors connected with the propellers run slower with increased torque, which is a good advantage when high-speed prime-movers, such as steam turbines, are employed."

The British master patent (as it is, because no other patent had been sealed for such a combination of electro-mechanical marine propulsion machinery) was No. 17248/1905, and the claim No. 1 in same reads as follows:

"In a method of propelling a vehicle or boat, comprising the combination of a prime-mover and electric generator, a motor in electrical connection therewith, and appropriate transmission gear (propeller shaft), the use of an electric generator having a revolving magnet or magnets and a stationary armature, the windings being arranged for polyphase alternating currents."

The invention was also granted in various countries, whilst America, Germany, Russia and Canada are still pending decision, and that is why I at present have no claim, financially speaking, on the system used in

American waters on the *Jupiter;* but even she must not "legally" come within the 3-mile limit, recognized by the law of this country protecting British patentees, without permission from the patentee.

And that is the reason why I must not use the Thomson-Houston Company's patent steam turbine, nor their patent electrical machinery without their permission in writing or by purchase. But it is equally right that, although holding the above patents for the construction of certain types of steam turbines and electrical machinery, they must not use my patented system of electrical ship propulsion, as installed on the *Jupiter,* in this or any other country in which my inventions have received lawful recognition.

In 1908 I further improved on the system by bringing out my invention No. 23396/1908 (British), Germany (No. DRP 230406/1909), France (No. 408561/1909), Italy (No. 105779/1909), Switzerland (No. 50610/1909), Belgium (in which the patents have been sealed), America (No. 525735), Russia, and several others are still pending. In this invention means were produced by which the screw propellers could be turned at variable speeds by generating variable-frequency alternating current at constant engine or turbine speed, and which is embodied in the large internal combustion railway locomotives now under construction on my system.

WILLIAM P. DURTNALL.

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## TURBINE DESIGN.

BY GERALD STONEY, F.R.S. *"Engineering," February* 13, 1914.

A very readable and concise description of principles of turbine design as applicable to land practice principally. Should be of particular interest to all desiring to know the essential basis of late turbine design.

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## AUXILIARIES.

### FIVE-STAGE CENTRIFUGAL PUMP.

An electrically-driven centrifugal pump is being fitted in the boiler house of the Stuart street generating station of the Manchester Corporation for feeding the boilers. It is of the high-lift pattern, and has been designed to deliver 27,500 gallons of water per hour as a normal load against a total head of 550 feet—240 pounds per square inch—and to deal with 30,000 gallons per hour as a continuous overload. The pump has five impellers, and both these and the guide passages are of phosphor-bronze. It is the product of Holden & Brooke, Limited, Manchester, and will be driven by a motor of Siemens Brothers' make, designed to develop 145 brake horsepower continuously, so as to allow an ample margin of power when working on the overload duty. The motor is arranged for a speed variation of between 1,300 and 1,500 revolutions per minute.

Fig. 2 is a sectional elevation of the pump. The overall dimensions of the complete plant are: Length, 8 feet 6¾ inches; width, 3 feet 8½ inches, and height, 4 feet 11¼ inches. The pump has 6-inch suction and delivery branches. The stuffing box on the suction end is of the water-sealed type, and is fed by water, which leaks past the periphery of the first impeller, thus ensuring a supply as free from grit as possible. The suction end bearing is of the ring-lubricated pattern, with a large oil well provided with an overflow to prevent overfilling, and has a renewable bush

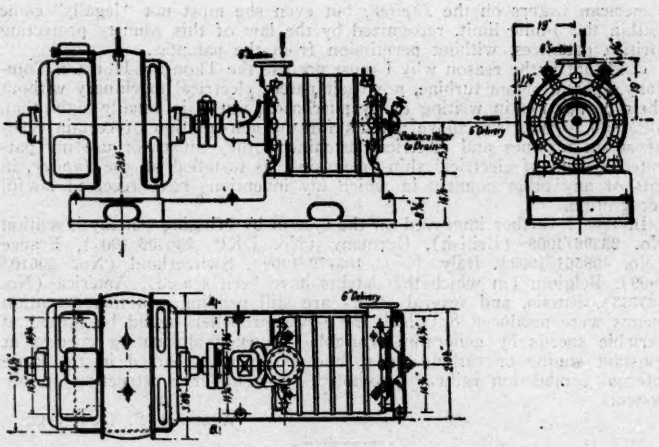

FIG. 1—FIVE-STAGE CENTRIFUGAL BOILER FEED PUMP.

of ample dimensions. The delivery end bearing is of an internal type, which dispenses with a stuffing box. It has a renewable bush, and to ensure coolness the water required for balancing the end thrust of the impellers is made to pass round the outside of this bush. As the bearing is made with a closed end, there can be no leakage of water along it, while any lubricant which may find its way out of the bearing only mixes with the balance water. In this way no lubricant becomes mixed with the water passing through the pump. This bearing is oiled by means of a screw-down lubricator, as shown. The main shaft is completely covered,

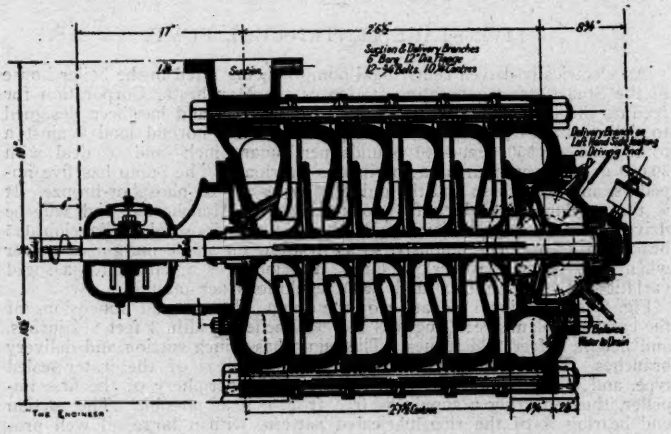

FIG. 2—FIVE-STAGE CENTRIFUGAL BOILER FEED PUMP.

partly by the impellers themselves and partly by gun-metal bushes, thus preventing corrosion.

The end thrust in this kind of pump is balanced hydraulically and automatically. This is done by means of the valve A—Fig. 2—which is carried on the shaft by means of arms, and is so constructed that it fits into a ring cast on the delivering side of the last impeller. This ring is of such a diameter that even if there were no pressure on the area within it, there would be an axial thrust on the impellers towards the delivery end. The load necessary to balance the thrust on the impellers is taken on the back of the last impeller, as this forms one wall of the balancing chamber C. The water for balancing passes from the periphery of the last impeller into C by way of the valve A. The outlet from C is controlled by a fixed and definite orifice D, by which the water is led away to the drain. In this arrangement any slight wear which takes place on the valve face is automatically taken up by a slight axial movement of the shaft, and no extra balance water is required as the outlet is controlled by an orifice independent of any running parts of the pump.—"The Engineer."

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## A TURBO-GENERATOR FOR SHIPBOARD USE.

Prepared from a paper submitted by FRANK E. FISHER, electrical designing engineer for the Diehl Mfg. Co.

Apparatus for use on shipboard in general and for use on United States war vessels in particular must be light in weight, compact in design, sturdy in construction, high in efficiency and capable of withstanding sudden and severe overloads.

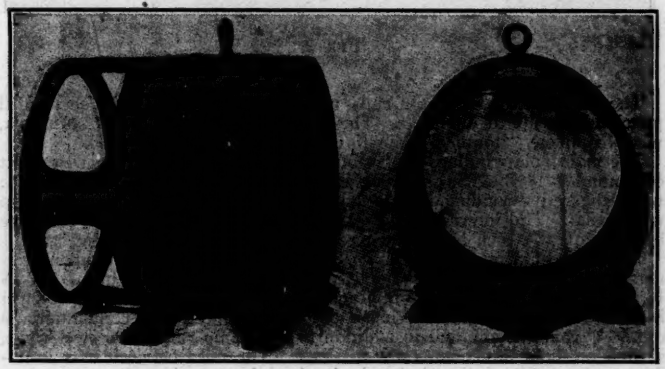

FIG. 1.

In choosing a generating unit for this service turbine drive offers many advantages, and in most instances is adopted. The high turbine speed reduces the size of the generator but makes mechanical construction, commutation and bearing problems more difficult.

The special features of a turbo-generator to meet this service are considered in this article and the particular machine is a 10-kw. 125-volt

compound-wound 3,600-r.p.m. generator, driven by a Terry steam turbine, operating under 200 pounds' steam pressure, with an exhaust of from 26 inches vacuum to 5 pounds' back pressure. This discussion is limited to the generator.

*Mechanical Construction.*—The yoke or frame of this machine is a steel casting, cylindrical in shape, with an extension on one end in which are openings through which the brush rigging and the commutator are readily inspected. (Fig. 1.) The particular shape of the yoke is such as to insure a smooth, homogeneous casting free from blow holes. The ends of the yoke are faced off at the same time that the pole seat cut is taken, thus insuring accurate concentricity of the poles, brush rigging and bearings. The interchangeable end shields, or bonnets, carry solid bronze bearing sleeves of the ring-oiling type. A ratio of bearing length to diameter of two and a half to one, proper clearance, and an oil ring of such size that it lubricates the bearing freely without throwing oil out of the housing, insure cool operation of the journals. A good grade of dynamo oil of medium weight is used.

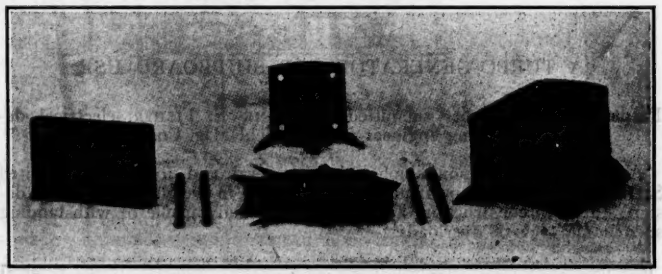

FIG. 2.

The main pole pieces are built up of carefully annealed sheet-steel punching assembled on rivets. The shape of the punchings is such that, when complete, "toothed" pole tips are obtained. (Fig. 2.)

The main pole windings are wound separately. The shunt coils consist of a number 18 B. & S. gage single cotton-covered wire wound in layers; while the series winding is built up of 1/32-inch by ¾ inch copper strap wound flat. The interpole pieces are of wrought iron and the coils are built up of 1/32-inch by 2¾-inch copper strap wound flat. (Fig. 3.)

The ratio of armature length to diameter is larger than common, giving a comparatively low peripheral speed. The core is built up of No. 28 B. & S. gage sheet steel carefully annealed after punching, and each sheet is given a coat of insulating japan before assembly. The commutator is of the V construction, and in addition to the clamp rings, two steel rings, shrunk on, hold the commutator in shape. The mica between bars is undercut about 1/64 inch. A brush density of approximately 30 ampères per square inch obtains at full load. A balance ring with drilled and tapped holes for receiving balancing weights is mounted at each end of the armature. The brush rigging is mounted on a rocker ring which is split in two parts and revolves about the bearing housing.

The form-wound armature coils are riveted and soldered to high commutator risers, giving a fan effect. Additional ventilation is obtained by means of longitudinal vent holes through the core, and ventilating spacers transverse to the shaft.

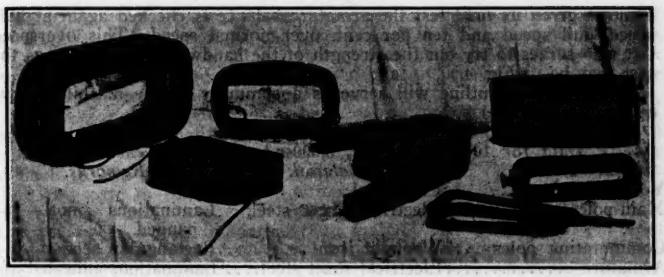

FIG. 3.

*Electrical Designs.*—The use of commutating poles, if properly designed, insures perfect commutation, which means a long-lived commutator. It also permits operation with a low ratio of field ampère turns to armature ampère turns, which means a small magnetic circuit and field winding resulting in light weight and small space. Commutating poles have a marked compounding effect, reducing the large number of series turns. A plain shunt-generator with a strong commutating pole winding can be flat compounded with a proper bush setting, but without series windings. However, parallel operation of the machines is impossible if good results are desired.

FIG. 4.

The strength of the commutating winding in the case of the generator under discussion is approximately 20 per cent. stronger than the armature, while the ratio of field strength to armature strength is about one to one.

The armature balance must receive particular attention for such a machine. It is usual to subject the armature to careful balance tests during various stages of its manufacture and after completion. After the ma-

chine is given its final test the armature balance is checked again at half speed, full speed and ten per cent. over normal speed. This overspeed test also serves to try out the strength of the bandwires or wedges, as the case may be. (Fig. 4.)

The following outline will serve as a summary of the machine above considered:

*Data for 10-kw., 125-volt, 3,600-r.p.m., Turbo-Generator.*

| *Part.* | *Material.* | *Remarks.* |
|---|---|---|
| Yoke | Cast steel. | |
| Main poles | Electrical sheet steel, | Laminations not japanned. |
| Commutating poles | Wrought iron | Rectangular section. |
| Armature | Electrical sheet steel | Laminations annealed after punching; insulated with japan. Armature diam. = 7½ inches. Peripheral speed = 7,080 feet per minute. |
| Band Wires | Phosphor Bronze on core. Steel end Bands. | |
| Commutator | Hard drawn-copper bars. | Bolted type. Diameter is 5 inches. Peripheral speed 4,720 feet per minute. |
| Shrink Rings | Nickel-steel | Two rings—one at outer end of commutator—one behind risers. |
| Brush Holders | Bronze and brass | Baylis reaction type. |
| Brushes | Carbon | Special turbo-generator brush, light in weight, graphitic in composition. |

Weight of generator..... 525 pounds.
Weight of set..... 1,350 pounds.

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## CARE OF THE ELECTRIC PLANT ON BOARD SHIP.

### FAULTS IN THE ARMATURES OF DYNAMOS—HOW TO FIND AND REMEDY THEM.

By Sidney F. Walker.

A fault is the term used by electrical men to describe a source of trouble; something that either stops the apparatus working or causes it to work badly. The armatures of all dynamos, whether used as generators or motors, are the seat of frequent troubles. Probably faults in armatures are more prevalent than faults in all of the other parts of the apparatus put together. It is the armature that is doing the work, that is subject to all the strains—both electrical and mechanical—that is subject to wide changes of temperature, and so on.

The principal faults found in dynamo armatures are:

(1) A break in the circuit of the armature.

(2) A connection between two adjacent coils, or between two adjacent wires of the armature.

(3) A connection between one or more of the coils upon the armature and the core.

Fault No. 1 is perhaps the most serious fault of all, because it stops the working of the machine. With faults Nos. 2 and 3 the machine may go on working for a certain time and may even be made to continue working under special conditions, providing that it is understood the convenience will have to be paid for later on.

A break or a disconnection in the armature may arise in the middle of one of the coils, though it is very rarely that such a thing happens. When such a fault does occur it is a very troublesome one, because it is so very difficult to find and because the coil has to be taken right off for repair. The only case in which, in the writer's experience, such a fault has occurred has been where either inferior wire has been employed, where there has been a weld in some part of the coil, or where a joint has been made in the wire with which the coil is wound. Modern dynamo makers are very careful indeed as to the quality of the wire which they employ, and as to the firms whose wire they use; but with the very best firms and with the greatest amount of care it may sometimes happen that there is a weld in the wire, somewhere in the length of an armature coil. In the process of wire drawing welds are necessary. Experienced wire drawers make then as few as possible; but it may happen that a wire breaks in going through one of the draw plates, and rather than sacrifice the whole coil a weld is made, as carefully as possible, and after the wire has been covered with two coatings of cotton, the cotton being varnished or protected by one of the insulating materials upon the market, or even when the wire has been enameled, according to the latest method of insulating wires for electric magnets, it is practically impossible for the man who winds the coils to know of the existence of the weld. If the weld is properly made, probably it is as good as the wire itself. It will introduce a very small resistance into the circuit, and its presence might be suspected, by very careful tests, from this cause.

On the other hand, welding tends to make the wire brittle just by the side of the weld, and it may happen later on, when that particular portion of the coil is exposed to a considerable amount of heat, possibly followed by contraction, and again followed by heating up, that the wire may break just beyond the weld. It may also happen that by accident, or through shortage of wire in the works, together with pressure for delivery, two pieces of wire are jointed together to form a coil. This, of course, should not be done; but under pressure things of the kind often are done, particularly when repairs to machines are executed. The jointing process has the same tendency to render the wire brittle just beyond, and the same thing may happen.

But by far the most common cases of disconnection occur at the commutator; particularly after repairs have been executed. It will be remembered that the adjacent ends of two armature coils are connected together, and to a segment of the commutator. Sometimes there are arms standing up from the commutator, sometimes the wires are embedded in slots in the segments. With either method it requires considerable care and skill to make sure that the joint is a good one. The slot in which the wires are to be held should be carefully cleaned, all grease thoroughly removed, the slot itself thoroughly tinned, the ends of the wires thoroughly tinned and the whole bedded together with solder.

Another possible cause of disconnection is that the ends of the coils may be broken off between the armature and the commutator. This is due to various causes that have not been thoroughly investigated.

Both a connection between two adjacent coils or between two adjacent wires of the same coil, and a connection between any coil and the iron core, will lead to heating of the armature and to sparking. It may be taken as an axiom that when sparking increases the condition of the armature should be suspected. It may be due to the causes mentioned in the article dealing with sparking; but if, after applying the remedies

mentioned there, sparking still continues, the armature should be carefully examined for heat. When the two wires, forming the ends of a coil, are connected together, that coil will heat up, and it will usually heat so badly that there is no question as to where the trouble is. When a coil is in connection with the iron core the heating will be less; but it will usually declare itself by heating up that particular coil to a certain extent and by increased sparking when that particular coil passes under the brush.

The cause of two wires forming the ends of a coil being connected together is very frequently at the commutator. If the commutator has been trued up, it may happen that very minute pieces of copper are left connecting two adjacent segments together. It has also frequently happened that when a commutator has been taken off to repair the armature and has been replaced, the two ends of one coil have been connected to the same segment of the commutator instead of the two ends of adjacent coils, as it should be.

The cause of a coil becoming connected to the core is usually a fault brought about during construction. The armature core is built up of a number of thin iron or mild-steel plates, from which pieces have been punched out, the gaps so made together forming the channels in which the wires are laid when the armature is wound. Unless great care is taken it may easily happen that very small pin points of iron or steel are left sticking up, and with the increase of temperature to which the machine is subject when it is running and the vibration to which the armature is also subject, these pin points may work their way through the insulation and make connection with the inner coil.

One connection between a coil and a core will not do a great deal of harm; it will merely give rise to heat, and possibly the machine may have to run a little faster; but two such connections may cause the machine to refuse to furnish its full pressure, unless driven at a very much higher speed than the normal; and it will lead to the development of a very large amount of heat in the coils between the two connections.

### HOW TO FIND THE FAULTS.

A good deal may be done toward finding all the faults by carefully observing the sparking at the commutator. With a break in the armature circuit no pressure will be furnished, but there will be a small spark when the disconnection comes round to the negative brush in the case of the generator and the positive brush in the case of the motor. It is not always easy to locate the position of the spark, and therefore some kind of test has to be applied. The surest method is to make another break in the armature by disconnecting two ends of adjacent coils from their segment of the commutator and carefully separating them. Then by testing round with a small battery and either a galvanometer or a small cell-testing voltmeter, the break will be easily located. Having formed a testing circuit of the battery and the galvanometer or voltmeter, one end of the circuit is connected to one of the wires that have been disconnected from the commutator, and the other wire is touched on each segment of the commutator in succession. The galvanometer or the voltmeter will give a deflection at each segment until the break is passed. The break, however, may be found without disconnecting, by the aid of a telephone and a battery.

When the position of the disconnection is found—if it does not appear to be at the commutator itself, as mentioned above—that coil will have to be taken off, and it will probably be found that there is a break inside. The coil which is short circuited, between which connection has been made either by a piece of copper between the commutator segments or the equivalent, needs no testing. It shows by the heat in the coil itself;

and the writer would like to give the practical hint that, after an armature has been repaired, if the commutator has been disconnected—even if only a portion of it has been—the armature should be run slowly, without any load, and the coils very carefully examined for heat. The heating will take place when the armature is run very slowly and without any connection to the outer circuit.

A connection between one of the coils and the core is also easily found by the aid of a battery and either a galvanometer or a testing voltmeter. A testing circuit is formed with the battery, the galvanometer or voltmeter and the two end wires. One of the end wires should be connected to the armature shaft, the brushes having been thrown back, and the other end wire touched on one of the commutator segments. If there is a connection between any of the coils and the core a deflection will be shown upon the galvanometer or voltmeter; and by passing the testing wire round the commutator, or, what amounts to the same thing, by holding the wire in one position and moving the armature slowly round, the position of the coil connection will be found by the fact that, with a galvanometer, the deflection will be reversed.—"International Marine Engineering."

## TERRY TURBINES DRIVING F. D. FANS ABROAD.

Terry turbines are being extensively used on Destroyers built by Yarrow in conjunction with Yarrow turbo-fans for forced draft. In this country Terry turbines have been used in conjunction with Sirocco, Sturtevant, Multivane and Keith fans, as well as with plate fans.

The following extract from "The Engineer" indicates the extensive use of Terry turbines for this service abroad.

Yarrow & Co., Limited, have already fitted five sets of these turbine fans in each of H. M. torpedo-boat destroyers *Archer* and *Attack*, and six sets in each of four torpedo-boat destroyers of the "L" class, two of which vessels are already delivered and have passed satisfactorily through their trials; the two remaining vessels will shortly run. They are also fitting six sets in each of three torpedo-boat destroyers of the "M" class, which have been ordered from them by the Admiralty, and six sets in each of two destroyers for the Imperial Japanese Navy now being built by them. On trials recently carried out with H.M.S. *Lark* and *Linnet*, it was found when developing about 9,000 horsepower in one stokehold, and burning oil fuel of the Texas quality, that it was quite easy to maintain an air pressure of 6½ inches water gage with two fans running.

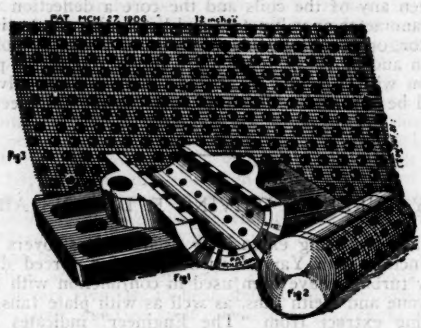
We understand that Yarrow & Co., Limited, have secured the license from the Terry Company, of Hartford, United States of America, for manufacturing this type of turbine, which is also extensively adopted in the American Navy.

## GRAPHITE LUBRICATION.

The employment of graphite as a lubricating medium for machinery is a matter which has often been discussed among engineers. The mixing of graphite with oil, a system tried for engine lubrication and other purposes, has not always been attended with such satisfactory results as are to be desired, but there can be no doubt that, properly applied, graphite ranks among the most efficient of all lubricants. Countless experiments have been made to unite the substance with various metals in a molten state, but without success, and another system of application has con-

sisted of graphite plugs inserted in bearing surfaces, but here the cost of drilling the holes for each separate plug and putting each individual one in its place has made the process too expensive for ordinary purposes.

In view of these circumstances it is with interest that we turn to examine a new method of utilizing graphite as a lubricant for white-metal bearings, particulars of which have been communicated to us by the vendors. This is known as the Randall graphite sheet lubricator and consists of a simple device in which solidified graphite cones of a slightly tapered formation are attached by hydraulic pressure to a fine copper-wire screen or fabric which may be cut to the shape and fitted with a minimum of trouble in any white-metal bearing in which it is intended to

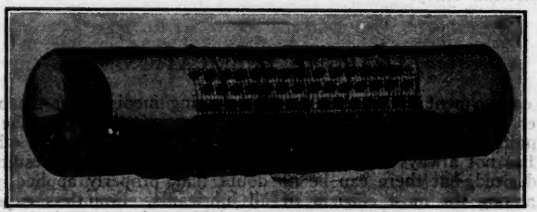


MATERIAL IN SHEET FORM ON THE MANDRIL AFTER CASTING INTO THE BEARING.

be used. A piece is cut from the sheet the full length of the bearing, wide enough to reach not quite to the top of the half bearing, as shown in the illustration, and it is important when fitting to so cut the sheet that the plugs or cones have the straight rows running lengthwise, so that the cones are staggered in the path of the rotating journal, and each part of the shaft in the bearing as it revolves will then be in contact with some of the graphite cones.

After cutting, the Randall graphite sheet lubricator can be shaped with the fingers to a half circle a little smaller than the journal, around which, or around a mandril usually employed for the purpose, it is fixed by means of fine copper wire, wound spirally.

The small or tapered ends of the cones should always be placed in con-



SHEET LUBRICATOR ON THE MANDRIL.

tact with the journal, so that the wire cloth will be embedded near the bottom of the box.

The mandril, with the Randall graphite sheet lubricator attached as described, is placed in the box and the white metal poured in in the usual way. The bearing box, mandril and sheet should all be warm so as not to chill the white metal too quickly. The metal will flow and set around the graphite cones, making them a part of the bearing.

The result is practically a self-lubricating bearing which requires only a few drops of oil at long intervals.

BEARING COMPLETED WITH SHEET LUBRICATOR CAST IN.

To finish the bearing the sides may either be scraped so that the journal will turn freely, or turned in a lathe, whichever is the general practice.

In oiling, it is advisable to cut down from 75 to 90 per cent. of the quantity ordinarily used on regular white-metal bearings. Too much oil has a tendency to wash loose particles of graphite out of the bearing.

In considering the first cost of the Randall graphite sheet lubricator it should be noted that in many cases it displaces sufficient white metal to pay for itself. It increases the life of bearings three to four times.

Results have shown that this lubricating device enables white-metal bearings to withstand heavy pressures and high speeds without heating, and it certainly, therefore, demands investigation and trial.

The economy claimed for these graphite sheets is from 75 to 90 per cent. of the usual expense of lubricating oils and greases, and although most users prefer to retain a slight percentage of oil, at most this need not be more than from 10 to 25 per cent.—"Shipbuilding and Shipping Record."

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## "AIREX" AUTOMATIC FEED-WATER AIR EXTRACTOR.

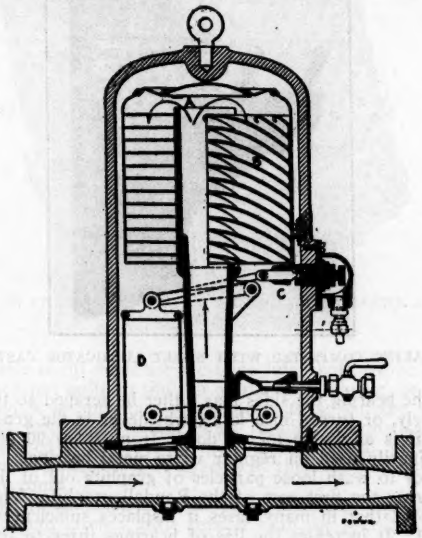
This device is designed to be installed in the boiler-feed line to eliminate air from the feed water. Water contains about 2 per cent. of air, which is carried to the boilers with the water and is contained in the steam leaving the boilers.

To facilitate the air extraction the apparatus is installed near the boiler

in the hot feed-water line, as hot water gives up the air easier than the cold water.

By eliminating the air corrosion of the boiler tubes is prevented, involving less repair cost and fewer shutdowns. Another advantage of the apparatus is that less air reaches the condenser and a higher vacuum will result. The increase of the vacuum, however, cannot be given in exact figures, as the conditions vary largely.

The automatic float valve, with which the air extractor is fitted, is placed so far below the highest point in the air vessel that when starting with atmospheric air in the air vessel and a leaky valve, the surface necessary for eliminating the air is not diminished. For this purpose, and also for removing the air as completely as possible, the feed water enters the apparatus at the highest point *A* of the air vessel, at a low velocity, and spreads in a thin film over the distributing surfaces *B*.




SECTION THROUGH "AIREX" AIR EXTRACTOR.

The discharge of the water into the distributing surfaces is placed so high above the air valve *C* that there are sufficient surfaces exposed for eliminating the air.

Air, gradually collecting in the air chamber, under a pressure equal to that in the boiler, causes the water level to sink until the float *D*, acting on the air valve *C* through suitable levers, opens the air valve and relieves the chamber of air. As the float is of the open type, it is not subjected to external pressure. It is so arranged that the valve only opens when the water level has sunk about 4 inches below the air valve. This arrangement prevents dirt and oil from clogging the air valve.

When the apparatus, which is manufactured by the General Condenser Co., 1239-1257 North Twelfth Street, Philadelphia, Pa., is in continuous use the water level will automatically remain at a constant height. By a special arrangement the oil and floating dirt can be drawn off.—"Power."

## STEAM WATER HEATER TEST IN COUNTERFLOW AND PARALLEL FLOW.

"Zeits des Vereines Deutcher Ingenieure," January 31, 1914, Comment in Journal A. S. M. E., March, 1914.—Test of water heater clearly indicated superiority of counterflow arrangement. Various data as to capacity and behavior under various conditions of steam pressure shown.

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## OIL ENGINES.

### THE MARINE DIESEL ENGINE.

By J. T. Milton.

Paper read before the Institution of Naval Architects, April 2, 1914.—This paper is reprinted in "Engineering" April 10, 1914. It is a comprehensive discussion of the Diesel engine as applied to marine work, and gives a complete account of the actual experience obtained with the latest Diesel engines afloat.

### THE MOTOR SHIP *FIONIA*.

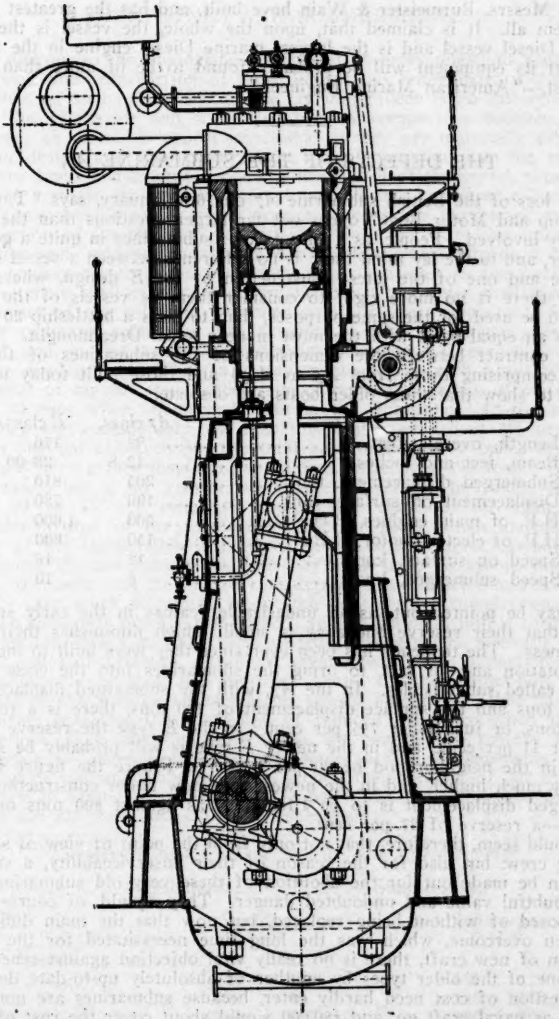
The Diesel-engine installation consists of two four-stroke main engines, each having six cylinders with a diameter of 740 mm. and a stroke of 1,100 mm. At 100 revolutions per minute these engines normally develop 2,000 I.H.P. each. It will be seen from the sectional drawing of the engine shown in Fig. 2 that the bed plate is constructed as in an ordinary steam engine. It is cast with an open crank pit, and running the whole length below the bed plate there is bolted a tray-shaped reservoir of welded steel plates in which the oil is collected. The supports are not cast as enclosed cases, as in the case of motors which have been formerly built by this firm, but are separately cast as A-shaped trestles, a trestle of this description being erected for each main bearing. These trestles are stiffly connected with the guides which are bolted to flanges on the sides of the standards. At the top on the front side the standard trestles are connected by intermediate pieces, the top being furnished with covers through which the piston rods can pass through a very simple stuffing box. Hence the whole of the front side of the engine as well as that portion of the back of the engine which lies below the guides, is therefore quite open and the engine is even more accessible than the ordinary steam engine. These openings are covered by easily-removable oiltight doors of steel plate. When these plates are fixed all the moving parts are totally enclosed in an oiltight case and are lubricated by forced lubrication.

The cylinder jackets are cast in two groups, each with three cylinders. They are fitted with feet which are placed on the above mentioned standard trestles. The cylinders are in this way accessible from below, which is a considerable improvement upon the previous motors, because this arrangement has a great advantage in that the cylinder oil dripping down from the cylinders cannot be mixed with the oil used for forced lubrication of the moving parts. The cylinder oil is gathered in a tray on the top of the standards and is carried to the filters through pipes. Nearly 60 per cent. of the cylinder oil can in this way be gathered and used again after the filtering. Large stay bolts are carried from the top of the cylinder jackets through the standards to the under side of the bed plate so that in all parts heavy tension strains are avoided. The crankshaft is

built up and, as usual, is bored out for admission of oil for forced lubrication. The connecting rods, crossheads, and piston rods are all constructed according to the usual marine practice of Messrs. Burmeister & Wain. The pistons are water cooled, for which purpose sea water is taken from the ordinary pressure pipe for the cooling water of the engines. This water is carried to and from the pistons by telescope pipes fitted with metallic packings. The alteration involved in cooling the pistons by water instead of oil has not been undertaken on account of difficulties in oil-cooled pistons, but because it would have been a difficult matter to have cooled so considerable a quantity of oil as that necessary to cool so large a piston.

The valve motion is the same as in previous similar motors, the only alteration being that the reversing is not carried out by a rotating air motor, but by a reversing machine of a similar type to the Browns gear. This is driven by the maneuvering air and works in a reliable and noiseless manner. Another point that may be noted is that the cam shaft is not driven by connecting rods as in the previous motors, as, instead of this, two large toothed wheels are used. This arrangement can be made more exact than the former one from a constructional point of view. The fuel pumps are driven by levers from one of the intermediate shafts to the cam-shaft gear, one fuel pump being employed for each cylinder. The compressors for injection air are driven directly from the end of the main shaft, these compressors being made by Messrs. Reavell & Co., of Ipswich, England. The plant differs from those used in the motor ships which Messrs. Burmeister & Wain have formerly built, as such vessels have compressors for twenty atmospheres driven from the auxiliary motors, having only one high-pressure pump compressing the air from twenty to sixty atmospheres. If this arrangement had been adopted in the motor ship *Fionia* with her 4,000-H.P. motors, it would have been necessary to increase the auxiliary motors to 300 H.P., and such a motor would have been too large for working the electric winches. With high-power units such as those on the *Fionia*, the better method is to arrange the compressors on the main engines themselves. The main engines are started by air at 25 atmospheres stored in two large air reservoirs, each with a volume of 800 cubic feet. This air can be taken from the compressors on the main engine, but for maneuvering purposes it is taken from an auxiliary Reavell compressor of 200 B.h.p. and can serve as a complete spare for one of the compressors on the main engines. In order to work this compressor and also to produce electricity for the auxiliary machinery, steering gear, winches, lighting, etc., two four-stroke Diesel engines, each normally developing 200 B.h.p. at 225 revolutions per minute, are employed. These motors are directly coupled to continuous-current dynamos of 220 volts pressure for working all the motors. The current for lighting is transformed to 110 volts. One motor is sufficient for working all the auxiliaries and winches, so that there is always one auxiliary Diesel engine as a spare.

The auxiliary machinery, which is all electrically driven, consists of two cooling water pumps which are evolvent pumps with vertical shafts, two sets of pumps for forced lubrication made as rotating wing pumps, and two sets of bilge sanitary and piston-cooling pumps constructed as plunger pumps and driven from the same crank shaft. This last mentioned pump draws the cooling water from the piston-cooling system and forces it overboard. In each case it is only necessary to have one of these pumps running at a time, so that the alternative pump can stand as a spare. Additional to the above installation there is a ballast pump made as a rotating pump, a syringe and wash-deck pump, a fresh-water pump, a fuel-oil pump, a refrigerating machine and a crude-oil motor for lighting the ship when it is in port and the winches are not working. This last-named pump can also be used for driving the small compressor for



the purpose of pumping up air in the flask for the auxiliary motor, in order that the Diesel engine can be started. In the front edge of the casing there are arranged two daily supply tanks, each able to take the necessary fuel oil for twelve hours, working on full load. These tanks are pumped up alternately, so that the oil has time to settle any water and impurities in it before it is used in the motors.

The motor ship *Fionia* described above, is the ninth large motor ship

which Messrs. Burmeister & Wain have built, and has the greatest power of them all. It is claimed that, upon the whole, the vessel is the most rapid Diesel vessel and is the largest marine Diesel engine in the world, so that its equipment will be probably found to be of more than usual interest.—"American Marine Engineer."

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## THE DEFECTS OF THE SUBMARINE *A7*.

The loss of the British submarine *A7* on 16th January, says "The Motor Ship and Motor Boat," opens out far larger questions than the mere tragedy involved. People, as a rule, talk of submarines in quite a general manner, and to the lay mind there is no difference between a vessel of the *A* type and one of the latest construction of the *E* design, whereas in reality there is no more right to consider them as vessels of the same type, to be used for the same purposes, than to class a battleship 20 years old on an equal basis with the most modern super-Dreadnought.

The contract between the dimensions of the submarines of the *A7* type (comprising those from *A5* to *A13*) and those built today is sufficient to show that these older boats are obsolete.

| | *A7* class. | *E* class. |
|---|---|---|
| Length, overall, feet | 99 | 176 |
| Beam, feet and inches | 12–8 | 23–00 |
| Submerged displacement, tons | 205 | 810 |
| Displacement on surface, tons | 190 | 720 |
| H.P. of main engines, B.H.P. | 500 | 1,600 |
| H.P. of electric motor, B.H.P. | 150 | 600 |
| Speed on surface, knots | 12 | 16 |
| Speed submerged, knots | 8 | 10 |

It may be pointed out, as an undesirable feature in the early submarines, that their reserve buoyancy is small, which diminishes their seaworthiness. The tendency has been ever since they were built to increase this flotation and, in fact, to bring the submarines into the class commonly called submersibles. In the *A7*, with the submerged displacement of 205 tons and the surface displacement of 190 tons, there is a reserve of 15 tons, or just under 7½ per cent. In the *E* type the reserve is 90 tons, or 11 per cent., and in the newer *F* type is will probably be somewhere in the neighborhood of 20 per cent. In France the figure is invariably much higher, and in the newest type now under construction the submerged displacement is to be 1,100 tons, as against 800 tons on the surface—a reserve of 27 per cent.

It would seem, therefore, that not only from the point of view of safety for the crew, but also for the reason of their unserviceability, a strong case can be made out for the abolition of these very old submarines of such doubtful value and undoubted danger. They should, of course, not be disposed of without being replaced, but now that the main difficulty has been overcome, which was the long time necessitated for the construction of new craft, there is no really vital objection against renewing every one of the older types by another of absolutely up-to-date design. The question of cost need hardly enter, because submarines are not expensive as naval craft go, and £80,000 would about cover the cost of the most modern *E* type. When that sum is compared with the annual expenditure of the larger naval vessels, and it is borne in mind that these submarines are capable of doing enormous damage, and perhaps are as important a weapon of defence as a Dreadnought, it would hardly appear to be an ill-advised or extravagant suggestion to recommend the construction of many more of these newer submarines within the course of the next year or so.—"The Steamship."

## SUBMARINES AND SUBMERSIBLES.

The disaster to the Submarine *A7* has raised many questions regarding the design and construction of these vessels. A contemporary states in an article on submarines that to the lay mind there is no difference between the submarine and a submersible. A comparison between these two types of vessels is almost impossible as they are materially different both in design and also construction. The submarine has the reserve buoyancy tanks constructed inside the main shell of the vessel, thus having only a single shell, whilst the submersible has her reserve buoyancy tanks constructed outside, giving greater space inside the vessel, and the construction of the tanks on the outside greatly increases the amount of buoyancy and incidentally forms a double skin, which affords a considerable amount of protection from damage by collision. In some cases the inner or main shell of the vessel remains intact, the outer shell absorbing the impact of the collision. Another point of great importance is that in the ordinary submarine the amount of reserve buoyancy is very limited, and, in fact, cannot exceed more than about 15 per cent. of the surface displacement. That is to say, a submarine with a surface displacement of say 200 tons cannot have a reserve buoyancy of more than about 30 tons when submerged. But in the Submarine *A7* the reserve buoyancy was considerably less than this, being only about 7½ per cent. of the surface displacement, the total reserve buoyancy being about 15 tons and her surface displacement being a little over 190 tons. But the submersible has a far greater percentage of reserve buoyancy, ranging from 30 to 35 per cent. of the surface displacement, being more than double that of the submarine. Another point which increases the safety of the submersible is that the tanks, both reserve buoyancy and fuel, being situated outside the inner or main hull, increasing the space inside, the main hull may be of considerably less diameter and is consequently very much stronger. About the only constructional resemblance between these two types is that in both vessels the main hull is of circular cross-section owing to this form offering the greatest strength in resisting external pressure. Another advantage of the submersible over the submarine is that the improved shape of the outer hull, when running on the surface, secures better speeds, but on the other hand, the submarine possesses the advantage of reduced resistance when running submerged. The following table shows the comparative powers for the two types, two vessels of almost the same surface displacements having been taken, the submarine having a surface displacement of 327 tons, the submersible having a surface of 334 tons, whilst the submerged displacements are 395 tons and 475 tons respectively:

*Submerged and Surface Speeds and Effective H.P.*

A. = Submarine. B. = Submersible.

| Speed. | *Surface.* E.H.P. A. | B. | *Submerged.* E.H.P. A. | B. |
|---|---|---|---|---|
| 8 | 115 | 40 | 103 | 118 |
| 9 | 200 | 60 | 145 | 170 |
| 10 | 340 | 85 | 200 | 228 |
| 11 | 520 | 120 | 360 | 290 |
| 12 | 615 | 165 | 340 | 370 |
| 13 | 620 | 220 | 455 | 465 |
| 13.3 | 645 | 242 | 495 | 495 |
| 14 | 815 | 295 | 635 | 590 |
| 14.5 | 940 | 360 | 740 | 660 |

It will be noticed that at a submerged speed of about 13.3 knots the effective horsepower required is exactly the same for both types of vessel, and after this speed the submarine requires more power than the submersible. But for surface speeds the submersible attains far greater speeds on the same effective horsepower.

The advantages of the submersible over the submarine are briefly as follows: (1) Reserve buoyancy and fuel tanks being carried outside the main hull, the latter may be smaller diameter and consequently stronger; (2) the tanks forming a second skin provide good protection against damage to the main hull; (3) improved shape of hull secures better speeds when running on the surface; (4) greatly increased capacity of the fuel tanks affords considerably greater radius of action; and (5) a large amount of buoyancy can be secured by blowing out the water by compressed air, which would doubtless be a great asset in the time of disaster.—"Shipbuilding and Shipping Record."

## ORDNANCE AND MINING.

### THE DAVIS IRON RECOIL GUN FOR AEROPLANES AND DIRIGIBLES.

The following extracts are taken from a circular prepared by G. C. Davison, of the Ordnance Development Co. of America, describing the Davis gun and the general subject of the offensive equipment that may be installed on air craft. This is a matter of general interest, and, while the claims made are those of the maker of the gun and not the user, the points brought up merit most careful consideration and attention.

#### SMALL CALIBER AEROPLANE GUNS.

Guns of small caliber, such that the shock due to recoil is not injurious, have found their place in aeroplanes. As compared with bomb dropping, such guns have two distinct advantages. By starting the projectile with a high velocity it will, within certain ranges, have a high striking velocity (about 1,000 f.s.). Flowing from this advantage, there is greater accuracy, due to the comparatively small amount of time required for the flight of the projectile.

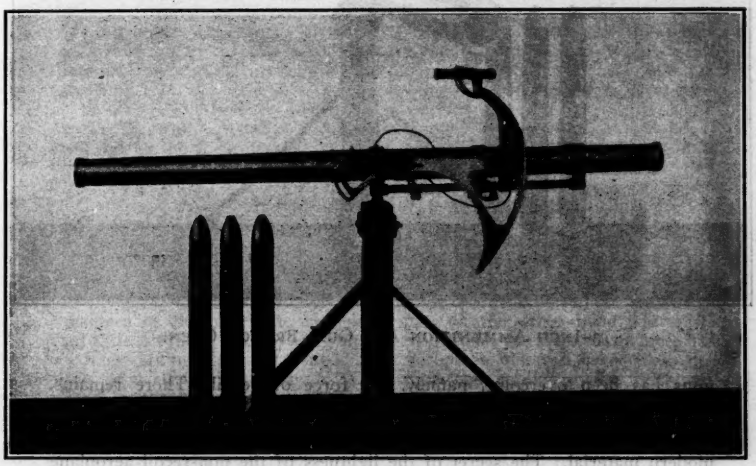
*Accuracy diminished by difficulty of aiming.*—But small guns necessarily have correspondingly short ranges. It may be found interesting to look into the question as to the probable limit of size of the usual type of gun which can be safely discharged from the usual type of military aeroplane of today. An investigation of the subject seems to indicate that the limit of the size of gun depends upon the velocity imparted to the projectile, the structural strength of the aeroplane, and the manner in which the gun is mounted. Merely for purposes of illustration, so as to obtain a fair idea as to the limit of size of gun, a moderate muzzle velocity of 1,000 feet per second has been assumed, with a powder pressure in the bore of five tons per square inch. From an examination of certain designs of aeroplanes, it seems that the gun could be so mounted that a recoil pressure of as much as 1,200 pounds could be sustained. Such pressure acting during the short interval of time in which it does, should not seriously interfere with the momentum of the aeroplane as a whole, but by its local action could bring severe stresses locally on the structure. If the gun be secured direct to some inelastic member of the aeroplane, without any recoil cylinder, or similar device, the pressure exerted on the plane would be large even for a small gun. Under the conditions

above given the limiting bore of a gun so secured would be about one-half inch.

However, on ships, it is the universal practice to absorb the energy of recoil by means of recoil cylinders, thus diminishing greatly the force acting on the structure during recoil. The greater the length allowed for recoil, the less the corresponding force of recoil. Assuming a comparatively light-weight gun, and allowing a recoil of about one-half the length of the gun, it is found that, by allowing a force of recoil of 1,200 pounds, the limiting size of gun would be about 1¼-inch bore, or about the size of the ordinary one pounder. The writer has no knowledge of any experiments having been made on an aeroplane with a gun of this size. The foregoing is merely intended as a statement of possibilities, and by no means recommends an armament of the kind described.

### THE NON-RECOIL GUN.

The invention of the Davis non-recoil gun, at once eliminates the limitation of size and power due to force of recoil. In this gun there is no force of recoil. The original conception of the inventor appears from his patents to have been a gun and a projectile carried by the aeroplane.



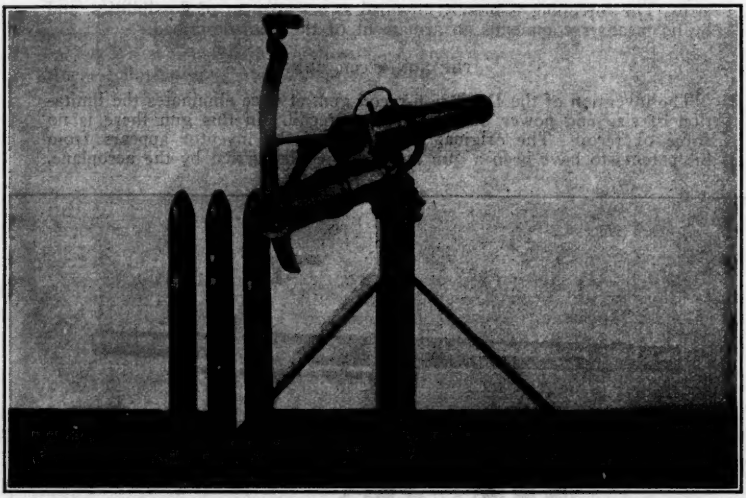
3-INCH AMMUNITION AND GUN, BREECH CLOSED.

The gun was so installed that on firing it was free to fly to the rear, while the projectile was discharged to the front. The gun, being free to move, was, of course, lost after the first shot. A further development of the invention contemplated a double gun, containing one powder charge and two projectiles. On the explosion of the powder, one projectile would be propelled toward the target, while the other would be thrown to the rear. The gun being of uniform bore,* had no surfaces on which the powder pressure could act longitudinally, and consequently, there would be no force of recoil. The latest development substitutes a mass of finely divided matter, such as metallic filings, or small lead shot, to be

*However, as this gun has no rifling, there may be some doubt as to whether the shot will fly true.

discharged from the breech of the gun, while the projectile is discharged from the muzzle. The dummy charge at once scatters, and after traveling a comparatively short distance, loses its velocity, and is not a menace to friends in the rear or below.† As worked out for practical use on an aeroplane, the non-recoil gun of moderate caliber has been made to act as a breech loader and uses fixed ammunition. The ammunition in outward appearance resembles that used in ordinary breech-loading guns. The design, as worked out, makes the aeroplane gun a light-weight, high-powered, quick-fire piece of ordnance.

Thus, with this invention the one great obstacle to large-size aeroplane



3-INCH AMMUNITION AND GUN, BREECH OPEN.

guns has been overcome, namely, the force of recoil. There remains, however, a certain limitation of weight. While this has not been entirely eliminated it has been wonderfully reduced by a departure from the usual methods of gun construction and by the adoption of the most modern material. The secret of the lightness of the non-recoil aeroplane gun lies in the use of high-grade, heat-treated alloy steel. In addition to the use of this high-grade steel a special combination of loading of smokeless powder has been adopted, so as to keep the powder pressure in the bore of the gun much less than ordinarily used, and at the same time give to the projectile a velocity of 1,000 f.s. or over. To illustrate the extent to which this system of gun construction may be carried it is necessary to go back to another type of gun invented by Commander Davis several years ago. This is the torpedo gun, which is carried in automobile torpedoes. While the gun was not of the non-recoil type, it was necessary, on account of the limited weight available, to make it as light as possible. As its life, when used in a torpedo, would be only one round, the gun was loaded so that the calculated powder pressure would just equal the elastic limit of the gun. When actually tested, this pressure

†This need not necessarily apply to biplanes.

was undoubtedly obtained, as shown by pressure gages, and also by measurement of the muzzle velocity of the projectile. Before making the test it was thought that the gun would be somewhat permanently strained, even if not destroyed, but on recovering the gun after the first test it was found to be in good condition, and was repeatedly fired during certain official tests covering a period of about two years. The gun fired an eight-inch shell weighing 250 pounds, while the weight of the gun itself was the same as that of the projectile. Thus on the same basis, an aeroplane, non-recoil (double gun) firing a 3-inch shell weighing 12 pounds, would itself weigh only 24 pounds. But, considering that the gun is to be fired by a man in contact with it, and that it is to be fired many times, it is built for aeroplane use with a large factor of safety.

Without going into details as regards pressures, stresses, etc., it may be stated that guns of this type may be given a factor of safety equal to that used in other types of modern ordnance, without exceeding the weight limitations imposed by mounting such guns upon aerial craft.

### TRAJECTORIES AND MARKSMANSHIP.

The conditions under which aeroplane guns are used are so entirely different from those pertaining to the land and naval gun that the ordinary formulae pertaining to trajectories do not apply directly. The factors which apply to the aeroplane gun, and which are different from those of other guns, are consequent upon the altitudes from which the aeroplane gun would be fired. A novel field for investigation is opened up in connection with the trajectories, or exterior ballistics of aeroplane guns. A certain amount of work has already been done in this connection, but as the formulae are somewhat complicated, and as there is little or no practical data from which constants can be determined, it is deemed inadvisable to here enter into a mathematical discussion of the problems. But, without going into mathematical investigation, a little consideration of the conditions will at once show from a common-sense point of view, some great advantages ballistically to be obtained by firing from high altitudes. Guns fired from the surface of the earth can only obtain comparatively long ranges by being fired at an angle of elevation, so that the projectile at approximately the middle part of its flight, attains a considerable altitude. The aeroplane gun, on the other hand, starts its projectile off from a comparatively great altitude, and in consequence, has a correspondingly greater range. The aeroplane gun can even, if so desired, be itself fired at an angle of elevation, thus adding to the height attained by the projectile. Not only is there an advantage in the form of long hitting range, due to the high altitude, but there is a concurrent advantage due to the action of gravity. There is a great amount of potential energy stored in a projectile when fired from an altitude of several thousand feet, and this energy is utilized to a certain extent in overcoming the resistance of the atmosphere and maintaining the velocity of the projectile. Assuming that a 3-inch projectile is fired from a height of 5,000 feet, the potential energy stored in the projectile due to this height, would be 60,000 foot pounds. If the gun be fired in a horizontal position, the range would be approximately 6,000 yards. The same gun, on the other hand, fired from the surface of the earth, would have an effective range of only about 2,000 yards.

Another great advantage to be enjoyed by aero artillery, as compared with land artillery, is the open view which can be obtained from aloft. In land battles where artillery participates there is frequently interference due to timber, hills, buildings, etc., which seriously hamper operations. A battery of aero artillery would never be restricted due to these causes. In the absence of actual data, it would be idle to attempt to predict the exact degree of accuracy obtainable with aeroplane guns. There are, it

is true, certain elements, such as height and speed, which tend to produce errors of shooting. It is thought, however, that the accuracy of observation possible should, in a large measure, compensate for these, and it is even possible that more accurate shooting can be done from the air than from a stationary gun on land.

### EARLY DEVELOPMENT AND TESTS.

The first demonstration of the principles involved in this invention was made with a 4-inch muzzle-loading gun 14 feet in length, weighing only 230 pounds. A number of ballistic tests were made with this gun firing two 4-inch projectiles each weighing 30 pounds. A muzzle velocity of 1,000 foot seconds was obtained with a maximum pressure in the bore of seven tons to the square inch. Delicate means were provided for measuring the recoil forces. This proved to be only a few ounces, and was no doubt due to a slight difference in the friction of the two projectiles.

This first experimental gun was followed by the breech-loading gun illustrated, which was constructed in order to test out the proposed design of breech mechanism, firing apparatus, etc. This gun has a bore of 3-inch diameter and over-all length of seven feet. The barrel is made of heat-treated vanadium steel with an elastic limit of over 100,000 pounds per square inch. The thickness of the walls in the middle of the gun is .625-inch, tapering towards each end to a thickness of .4 inch. The barrel is divided into two parts. One part is so mounted as to be handled very much like the breech plug of an ordinary gun. The two parts are connected together by means of an interrupted screw thread. This arrangement permits of the loading of the gun with fixed ammunition in a manner exactly similar to that used with the usual type of breech-loading guns. The ammunition is made up with the cartridge case having at one end a projectile, then the powder, and at the rear end the dummy projectile, composed of bird shot, held in place by suitable wads. Ignition of the powder is obtained by means of an electric primer located in the middle of the length of the cartridge case, the current being conveyed to this primer by an electric circuit commonly used in the electric firing of guns. It would be possible, also, to use mechanical firing, if desired. As shown in the photograph, the gun was loosely mounted on a light wooden support. When firing with the full charge this frail support was in no way disturbed, thus demonstrating the absence of recoil. The muzzle velocity obtained with this gun was a little over a thousand feet per second. The total weight of the gun complete, including firing gear, arm and operating mechanism, is 175 pounds. The weight of loaded cartridge is 30 pounds.

The tests of this gun demonstrated the success of the breech mechanism and firing apparatus and at the same time pointed the way to improvements in details. The elevating, training and sighting gear fitted were of a simple and elementary character designed only to secure certain data required for the design of corresponding parts for a practical and effective aeroplane installation. The illustration, therefore, while it gives a correct idea of the appearance of the gun proper, does not correctly represent the above-mentioned mechanisms in their final form.

### THE VALUE AND USES OF FLYING ARTILLERY.

For operations against troops in the field the great speed and mobility of the aeroplane render it an ideal mount for artillery. In addition, its practical invulnerability, the impossibility of concealing troops from its vision or protecting them from the effects of its fire are special advantages of the highest importance. As it will not be feasible to provide

protection from such fire, a great volume of fire will be of more importance than a great destructive effect from each shot. These conditions indicate that the guns used for this purpose will be of the rapid-fire breech-loading pattern and of moderate caliber, probably not in excess of 3 inches. Such an equipment will make it possible to carry a good supply of ammunition, thus enabling each aeroplane to remain in action for a substantial period of time. The great speed of the machine will shorten the time of absence from the firing line while the plane is replenishing its ammunition supply. Guns of this size and type would also prove useful in attacking arsenals, docks, magazines, and other similar structures located within fortified areas, to say nothing of sea coast and other fixed defences of similar character.

For naval purposes, a 3-inch gun of this type will be exceedingly effective against all unarmored ships, such as torpedo-boat destroyers, transports, mining ships, colliers, etc., and in addition, the gun fulfills a long-felt want in that is offers a method of attack on submarines. This type of vessel is entirely immune from ordinary gun fire, and up to the present time no method of defending against its attack has been developed. While, as is well known, an observer at or near the surface of the water cannot detect the presence of a submarine, it is also true that this invisibility is lost to a considerable extent when the observer is at a considerable height above the surface of the sea and nearly vertically over the vessel.

Still another field of usefulness for the moderate-caliber gun mounted upon an aeroplane would appear to lie in the attack upon dirigibles, in which case the use of a special type of projectile is indicated.

The larger-caliber guns would prove very effective, not only for attack on ships with horizontal armor, but also for attack upon fortifications, arsenals, docks, canal locks, magazines and similar structures. As has been previously pointed out, the larger-caliber guns will probably be of the single-shot type when mounted upon aeroplanes. When mounted upon dirigibles, however, the case will be different, as the great lifting power here available will permit of the use of guns designed for repeated firing. Obviously, a dirigible so armed, say with a battery of 8-inch guns, would be an exceedingly formidable weapon of war with a wide field of usefulness for both military and naval purposes.

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## THE MINE DEFENCE OF HARBORS.

By Capt. Paul D. Bunker, C.A.C., in the March–April, 1914, number of "Journal of the United States Artillery," is a comprehensive article on the subject of mining. It treats of the history of mining and gives descriptions of the various developments up to present practice. An article that should be read by all interested in the subject of mines and mining. The article is copiously illustrated.

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## GYROSCOPE. GARDNER'S.

"Engineering," February, 1913, describes a new form of gyroscope for torpedoes. The object of improvement is to secure runs free from precession.

## AIRSHIPS.

CAUSES OF ACCIDENTS, by Ing. Enrico Farlani, translated from "La Lettura," appears in "Engineering," February 13, 1914. Complete and detailed practical discussion of this matter is presented and specially recommended to those interested in aviation.

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## GREEN AEROPLANE ENGINE.

This engine is the product of the Green Engine Co., Ltd., Twickenham. Eng., and is fully described in its construction operation and performance in the January 30, 1914, number of "Engineering." Readers interested in aeroplane machinery will find it interesting to read this description.

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## AIR CRAFT ON THE DEFENSIVE AND OFFENSIVE.

### A REVIEW OF RECENT EXPERIMENTS.

France is zealously striving to adapt the flying machine to aggressive warfare. Among others, there were begun at the flying field at Buc in October, 1913, tests before a military committee with the so-called "Air Craft Hunter" of Lieut. Mailfert. The flying machine, which was already exhibited at the last Aeronautic Exhibition in Paris, is a Nieuport monoplane of 160 horsepower. It contains in its *fuselage* a shot-sieve or a projectile thrower, an arrangement which, when the pilot sets it in action, strews the underlying ground with two thousand shot. An aiming arrangement is said to be at hand. On the flying field a flying machine was represented to actual scale by means of a disc. Lieut. Mailfert rose in his machine to a height of 2,000 meters and flooded the target with his projectiles. The disc was riddled with many bullets. Then free balloons were released, against which also Mailfert brought his shot sprinkler into action. He succeeded in causing two balloons to sink.

Another kind of aggressive flying machine, the so-called "Zeppelin-Hunter," was put to trial last summer, according to the Paris "Journal." With its own weight reduced to a minimum, it had to carry an armor of 60 to 70 kilograms to protect both the pilot and the more vulnerable parts of the machine. It is intended to provide the flying machine with a light machine gun, which can perforate the Zeppelin hulls with rapid-fire shot and ignite the gas.

In January, 1914, an experiment took place also in Villacoublay, with a new armored flying machine, which was built and provided with a rapid-fire gun, according to the directions of the heads of military aeronautics. The ordnance, presumably a machine gun, is claimed to have signal striking capacity. The flight, however, which the pilot, Espaner, effected with the new aeroplane, was not very fortunate.

The French papers emphasize that in future all French military air craft, except the school machines, will be armored to protect them against hostile fire. They may be divided into the following kinds:

1. Armored monoplane for reconnoitering artillery and cavalry. Minimum velocity, 120 kilometers per hour.

2. Armored biplane for reconnoissance work of the general staff at great distances. Minimum velocity, 100 kilometers per hour.

3. Armored biplane which, provided with machine guns, etc., shall destroy hostile aeroplanes and airships. Minimum velocity, 120 kilometers per hour.

4. Large aeroplanes with more seats, for special service.

The Paris Aeronautic Exhibition last fall showed a special number of armored aeroplanes. The armor protection extended not only to the pilot and observer, but also to the provisions, the rudders, the measuring instruments and especially the propulsion system. The aeroplanes not intended for aggressive warfare were for the most part provided with wireless apparatus.

The improvement in the possibility of observation was shown at Villacoublay late in November, 1913, on a new kind of aeroplane in which the seat for the observer is placed forward, directly under the screw propeller. Deputy Girod, piloted by Gobé, flew over the field several times and made effective observations. The capacity for observations from this position is without doubt greater, still the observer in this advanced position is more exposed to hostile fire, and above all is incapable of taking charge of the controls of the aeroplane in case of failure of the pilot.

At Farnborough in England last year a trial of sharp-shooting from aeroplanes was held. A military biplane was provided with a machine gun and ammunition; this gave, from an elevation of 120 meters, 20 shots directly downward. As to its striking capacity and effect, nothing has been reported; the recoil from the shot was without notable influence on the equipoise of the aeroplane.

Also a Birmingham society instituted at Bisley sharp-shooting at an aeroplane with a machine gun. The 26-pound, air-cooled machine gun invented by Col. Lewis was fixed on a tripod on the aeroplane framing; on a tripod was also placed the movable seat for the protection of the aviator. Out of thirty shots at a target space of 20 feet, 11 hits were scored. From what elevation the aeroplane fired has, however, not been reported, nor the velocity of the air craft, nor the kind of target arrangement.

A sighting arrangement for machine guns on air craft has lately been patented. The machine gun is fixed between the legs of the observer sitting forward and having charge of the gun. The aviator sits elevated behind the observer. The fore part of the aeroplane is covered with a hood which carries on top a notch and sight. When the target is in sight the observer calls to the pilot and adjusts the sighting arrangement for distance and altitude. The aviator directs the aeroplane at the target by use of the rudders in conjunction with the notch and sight. When the observer perceives that the direction coincides he promptly fires.

An aeroplane projectile with rear rudder has been made in England. A tube extends lengthwise through the middle of the cylindrical projectile, filled with explosive. It carries, in front, trigger pins and percussion caps separated by an interspace, and having well to the rear an ignition charge. Both parts are secured by bolts fixed to the aeroplane frame, and, through the projectile, interlock in the tube, and so protect the aeroplane from destruction by its own discharge due to hard landing, shocks, etc. The part of the tube extending to the rear of the projectile carries the tail rudder. The projectiles hang on two hooks perpendicular to the floor of the aeroplane frame. When the projectile is to be thrown, the aviator establishes a special relation between the measured altitude and the velocity of flight, aims at the target, and, by means of a hand lever, releases the supporting arrangement of the projectile. The projectile, now free, sets itself point downward by means of its tail rudder. Since the two safety bolts have remained behind in the aeroplane, the igniter is in working condition, and when the projectile strikes, the trigger pin pierces the percussion cap, ignites the ignition charge, and this sets off the explosive charge of the projectile.

Shooting at aeroplanes was carried on last year in the camp at Chalons. A kite seven meters in width by six meters in length, represented an aeroplane and was towed by an automobile and wire cable with a velocity of 60 kilometers per hour. It was fired at by two squads of infantry, of fifty men each, at a distance of 800 to 900 meters. Out of nine hundred shots (nine cartridges for each man) twelve hits were scored on the kite, that is 1.3 for each man. A second shooting, with the same number of men, at a small kite, measuring 1.5 meters broad by 1 meter long, gave 0.5 shot per man, but at a distance not reported. Machine guns were trained against the same target, but with less effect.

A Spanish aeroplane was hit last November by Moroccans. Capt. Barreiro and Lieut. Gil-Rios undertook, in a reconnoissance on a military biplane, to observe the position of the insurgents. In this they were subject to fire by a group of hostile troops which had got footing on Conical Mountain. The lieutenant was grievously injured by two shots, the captain was wounded by one shot. It was possible for the officers, in spite of their wounds, to fly back again to their own camp—a proof, therefore, that even effective fire does not always put an aeroplane out of commission, and that an energetic aviator, even in spite of his wounds, will often carry out his orders.—"Scientific American" from "Kriegstechnische Zeitschrift."

## EMPIRICAL FORMULA FOR THE WEIGHT OF STEAM PASSING THROUGH A VENTURI TUBE.

JOHN B. GRUMBEIN, B.S. ME., GRADUATE STUDENT.

The investigation of which this formula is one of the results was developed after a year's work in the laboratories of Sibley College. The formula, as here given for the Venturi tube, is new, although a similar formula with a different coefficient was stated by Professor Rankine as applying to the flow of steam through orifices. He deduced it from the work of R. D. Napier.

The Venturi meter has so long been recognized as an instrument of precision for the measurement of the flow of water that it is difficult to see why its use has not been more general for the measurement of the flow of steam. If made of normal dimensions it is difficult to insert in a pipe line because of its length, and this may be largely overcome by increasing the throat diameter and thereby maintaining the angle of the diverging nozzle as less than ten degrees. This design permits the use of a short tube.

Professor Rankine proposed the following formula for the flow of steam through an orifice when the final pressure was greater than $\frac{3}{4}$ of the initial:

$$W = A\frac{P_2}{42}\left(\frac{3(P_1 - P_2)}{2P_2}\right)^{1/2}$$

in which

$A$ is the area of the orifice in sq. inches.

$P_1$ is the pressure in the reservoir, pounds per sq. inch absolute, and

$P_2$ is the pressure of the atmosphere, pounds per sq. inch absolute.

The following diagram gives the results of a series of actual experiments on the Venturi meter as compared with the calculated results from the Rankine formula for the flow of steam through orifices. In that diagram the dotted line corresponds to zero error, the full line to the results obtained by Rankine's formula. These results were 4.9 per cent greater than indicated by Rankine's formula and the experiments indicated that the coefficient, 42, employed by Rankine for orifices, should be replaced by 40 for the Venturi tube.

The experiments prove that the Rankine formula with the proper coefficient applies to the Venturi meter. For this purpose the formula is conveniently stated as follows:

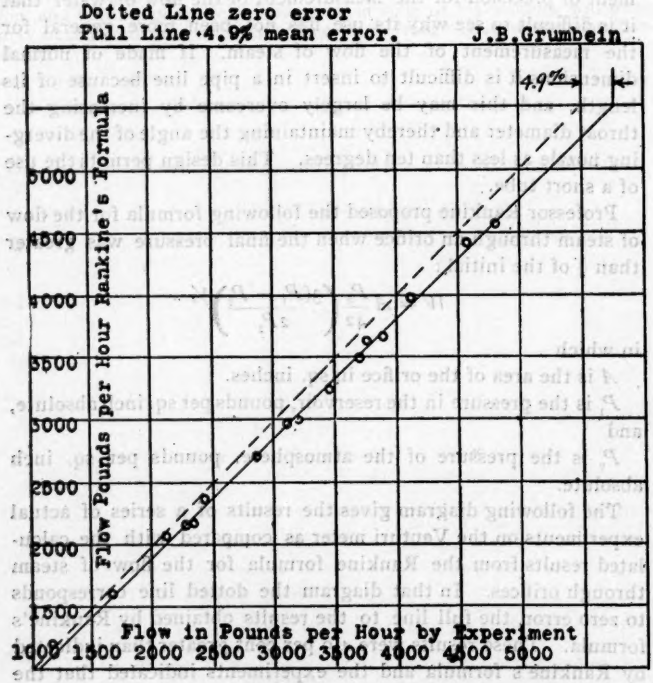
$$W = a_t p_t \cdot .03062 \left(\frac{p_e - p_t}{p_t}\right)^{1/2}$$

in which $W$ = pounds per second.

$a_t$ = area of the throat in square inches.

$p_e$ = pressure entering meter, pounds per square inch absolute.

$p_t$ = pressure in the throat, pounds per square inch absolute.




Curve showing the application of Rankine's Formula for the flow of steam through an orifice to the flow of steam through a Venturi meter.

# NAVAL VESSELS.

## UNITED STATES.

### UNITED STATES NAVAL VESSELS UNDER CONSTRUCTION.

| No. | Vessel. | Building yard. | Engines. | No. shafts. | Speed, knots. | Percentage machinery completed 1914. | | Hull com. May 1, 1914. |
|---|---|---|---|---|---|---|---|---|
| | | | | | | Apr. 1 | May 1 | Per ct. |
| | BATTLESHIPS: | | | | | | | |
| 36 | *Nevada* | Fore River S. Co. | Curtis turbine | 2 | 20.5 | 69.30 | 71.95 | 63.9 |
| 37 | *Oklahoma* | New York S. Co. | Reciprocating | 2 | 20.5 | 69.40 | 71.28 | 67.2 |
| 38 | *Pennsylvania* | Newport News Co. | Cur. trb. grd. cr. | 4 | 21 | 7.17 | 13.02 | 29.1 |
| 39 | ..... | Navy Yard, N. Y. | Pars. trb. grd. cr. | 4 | 21 | ... | ... | 11.5 |
| | DESTROYERS: | | | | | | | |
| 45 | *Downes* | New York S. Co. | Curtis trb. & rec. | 2 | 29 | 95.70 | 95.83 | 95.3 |
| 51 | *O'Brien* | Wm. Cramp & Sons | Cramp trb. & rec. | 2 | 29 | 64.90 | 70.11 | 69.2 |
| 52 | *Nicholson* | Wm. Cramp & Sons | Cramp trb. & rec. | 2 | 29 | 61.70 | 68.63 | 60.5 |
| 53 | *Winslow* | Wm. Cramp & Sons | Cramp trb. & rec. | 2 | 29 | 57.75 | 66.48 | 54.4 |
| 54 | *McDougal* | Bath Iron Works | Pars. trb. & rec. | 2 | 30.5 | 91.42 | 97.20 | 89.3 |
| 55 | *Cushing* | Fore River S. Co. | Curtis trb. grd. cr. | 2 | 29 | 56.99 | 61.14 | 41.5 |
| 56 | *Ericsson* | New York S. Co. | Pars. trb. & rec. | 2 | 29 | 51.19 | 58.22 | 61.6 |
| 57 | *Tucker* | Fore River S. Co. | Curtis trb. grd. cr. | 2 | 29.5 | 7.93 | 12.38 | 9.4 |
| 58 | *Conyngham* | Wm. Cramp & Sons | Pars. trb. grd. cr. | 2 | 29.5 | 0.59 | 3.80 | 6.8 |
| 59 | *Porter* | Wm. Cramp & Sons | Pars. trb. grd. cr. | 2 | 29.5 | 0.59 | 3.83 | 6.7 |
| 60 | *Wadsworth* | Bath Iron Works | Pars. trb. gearing. | 2 | 30 | 17.00 | 21.34 | 17.5 |
| 61 | *Jacob Jones* | New York S. Co. | Pars. trb. grd. cr. | 2 | 29.5 | 4.38 | 9.87 | 9.8 |
| 62 | *Wainwright* | New York S. Co. | Pars. trb. grd. cr. | 2 | 29.5 | 4.38 | 9.87 | 9.8 |
| | FUEL SHIPS. | | | | | | | |
| 13 | *Kanawha* | Navy Yard, Mare Isl'd. | Reciprocating | 2 | 14 | 35.90 | 37.80 | 41.1 |
| *14 | *Maumee* | Navy Yard, Mare Isl'd. | Diesel | 2 | 14 | ... | ... | 18.6 |
| | SUBMARINES: | | | | | | | |
| 31 | *G-3* | Navy Yard, N. Y. | Diesel-Sulzer | 2 | 14 | 63.30 | 75.00 | 81.6 |
| 34 | *K-3* | Union Iron Works | Diesel | 2 | 14 | 98.71 | 99.81 | 94.0 |
| 35 | *K-4* | Seattle Con. & D. D. Co | Diesel | 2 | 14 | 97.50 | 98.55 | 93.9 |
| 36 | *K-5* | Fore River S. Co. | Diesel | 2 | 14 | 97.99 | 98.48 | 92.8 |
| 37 | *K-6* | Fore River S. Co. | Diesel | 2 | 14 | 97.81 | 98.48 | 92.8 |
| 38 | *K-7* | Union Iron Works | Diesel | 2 | 14 | 90.75 | 93.28 | 88.9 |
| 39 | *K-8* | Union Iron Works | Diesel | 2 | 14 | 88.51 | 92.66 | 88.4 |
| 40 | *L-1* | Fore River S. Co. | Diesel | 2 | 14 | 35.58 | 67.54 | 30.3 |
| 41 | *L-2* | Fore River S. Co. | Diesel | 2 | 14 | 34.92 | 65.43 | 30.3 |
| 42 | *L-3* | Fore River S. Co. | Diesel | 2 | 14 | 32.08 | 44.22 | 30.3 |
| 43 | *L-4* | Fore River S. Co. | Diesel | 2 | 14 | 32.08 | 35.72 | 30.2 |
| 44 | *L-5* | Lake T. B. Co. | Diesel | 2 | 14 | ... | ... | 8.1 |
| 45 | *L-6* | Lake T. B. Co. | Diesel | 2 | 14 | ... | ... | ... |
| 46 | *L-7* | Lake T. B. Co. | Diesel | 2 | 14 | ... | ... | ... |
| 47 | *M-1* | Fore River S. Co. | Diesel | 2 | 14 | 29.07 | 53.75 | 20.8 |
| 48 | *L-8* | Navy Yard Portsmouth. | Diesel | 2 | 14 | ... | ... | ... |
| 49 | *L-9* | Fore River S. Co. | Diesel | 2 | 14 | ... | ... | 1.9 |
| 50 | *L-10* | Fore River S. Co. | Diesel | 2 | 14 | ... | ... | 1.9 |
| | SUBMARINE TENDERS: | | | | | | | |
| 1 | *Fulton* | Fore River S. Co. | Diesel | 1 | 12.25 | 68.78 | 74.93 | 62.3 |
| 2 | *Bushnell* | Seattle Con. & D. D. Co. | Parsons trb. with gearing. | 1 | 14 | 8.19 | 13.70 | 33.6 |
| | DESTROYER TENDER: | | | | | | | |
| 2 | *Melville* | New York S. Co. | Parsons turbine with gearing. | 1 | 15 | 5.80 | 13.84 | 47.3 |
| | TRANSPORT: | | | | | | | |
| 1 | ..... | Navy Yard, Phila. | Reciprocating | 2 | 14 | ... | ... | ... |
| | SUPPLY SHIP: | | | | | | | |
| 1 | ..... | Navy Yard, Boston. | Reciprocating | 2 | 14 | ... | ... | ... |

* Engines being built at Navy Yard, New York.

## SHIPS OF WAR.

We have found in the "Taschen Buch" of 1914 a table giving the general arrangement of the *Ersatz-Worth, Queen Elizabeth, Normandie* and *Pennsylvania* which are here reproduced.

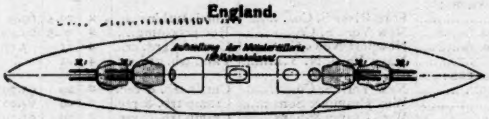
The first two vessels each carry eight guns of 15.2-inch caliber in four turrets, two at each end of the vessel. The third has three quadruple turrets with twelve 13.5-inch guns. The fourth battleship carries twelve 14-inch guns in four triple turrets.

This table represents the proposed armament of the most modern bat-

**Deutschland.**

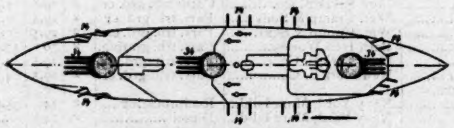
Ersatz Woerth. Breitseite: 8-31, 8-15
(Bewilligung erster Baurate 1913.)

**England.**



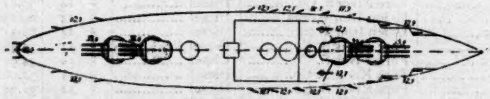
Queen Elizabeth. Breitseite: 8-38.1, 8-15.2.
(Bewilligung erster Baurate 1912.)

**Frankreich.**



Normandie. Breitseite: 12-34, 12-14.
(Bewilligung erster Baurate 1913.)

**Vereinigte Staaten.**



Pennsylvania. Breitseite: 12-35.6, 11-12.7
(Bewilligung erster Baurate 1912.)

tleships and offers a superb means of making comparisons of vessels armed with 8, 10 and 12 large guns.

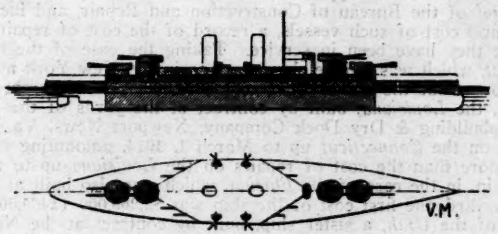
Is it preferable to provide a vessel with 8, 10 or 12 pieces? One need not hesitate to reply that the ideal arrangement is to place the largest number of the largest guns that can be accommodated. It is on account of this that of the four vessels we prefer the representative of the United States, armed with twelve 14-inch guns in four triple turrets. This offers the maximum offensive sector and at the same time four firing points. It also offers a very effective fire from forward.

It may be said that the employment of 340-mm. guns in the three quadruple turrets, as on the *Normandie,* is quite admissible. We do not deny that this arrangement offers the advantage of not having the tur-

rets superposed one over the other; but the center turret has a defective arc of fire, and there is also danger that the 340-mm. gun is too far below the 380-mm. from the view point of penetration.

Therefore we prefer the 356-mm. guns and the disposition used by the United States. In case we should consider an increase of caliber especially advantageous, we would consent to reduce the number of guns to eight, as in the English and German vessels.

It is a weighty question to consider whether we will not have to decrease the speed and increase the draught to more than 9 meters in order to secure greater displacements. If it is practicable, we would not hesitate to call for a tonnage capable of carrying four triple turrets of 380 mm. In case this is impossible we would endeavor to secure ten guns of 380 mm. in four turrets, the two lower turrets triple and the two upper ones double. In case this displacement is without bounds we should be satisfied with nine, and finally with only eight, as on the *Worth* and *Queen Elizabeth*.




"QUEEN ELIZABETH."

The more it is considered the more necessary it appears to take pains to increase the defensive qualities, not so much on account of the gun, but on account of the torpedo.

It is well to search for auxiliary means of defense against the torpedo. This defense is insufficient, and we are forced at last to consider armor for the underwater body of the vessel as a defense against this submarine engine. It will be necessary to reinforce the water-line armor belt with a plating of at least 10 c.m. and to complete the protection of the underwater body by two inner skins placed 2 and 4 meters, respectively, from the outer plating. Such an arrangement will have the effect of greatly reducing the volume of water that may enter in case of damage. This, again, adds to the displacement and makes it necessary to give super-dreadnoughts a draught of 9 meters.

The idea of armoring the underwater body is not new. In the work of foreign vessels published in 1913, one finds the above photograph.

With this new conception of naval vessels with a draught approaching 9 meters it will be necessary to vary the elements of length and breadth in a manner that will not cause a decrease in speed while using the same power. We may perhaps arrive at an enormous length, which may be limited by the weight of the hull, and maneuvering is rendered more difficult. It will be necessary to make special arrangements to secure better maneuvering qualities.

We may, for example, install a rudder forward or additional rudders aft, and, finally, the form of vessel may be modified to secure a better flow of water to the propellers. Just how this may be done we will not say; our role is more modest. We desire only to show that the idea of having a completely armored underwater body is again springing into life. Due to the length, to underwater armor, and to an efficient watertight

subdivision, a vessel will be able to resist numerous shot and, at the same time, the attack of numerous torpedoes and the assault of several submarines. It will not be at the mercy of one small accident.—"La Vie Maritime."

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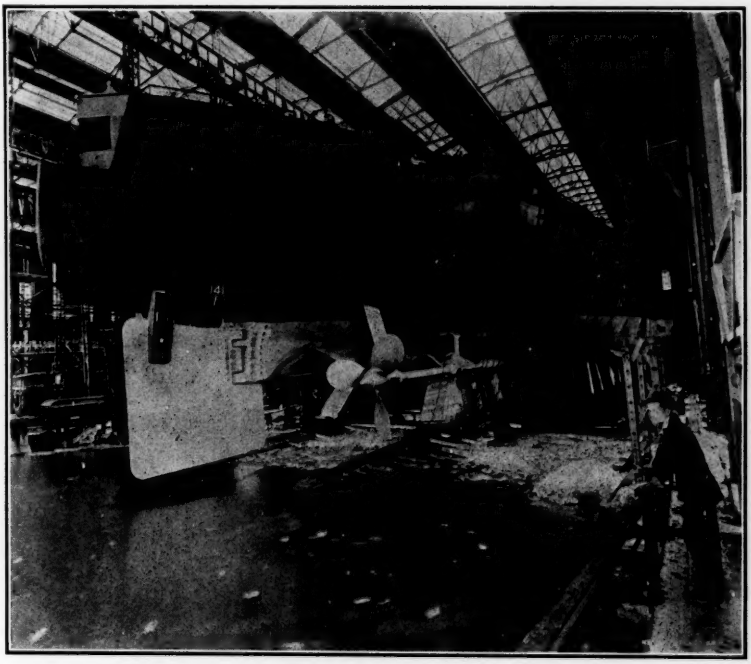
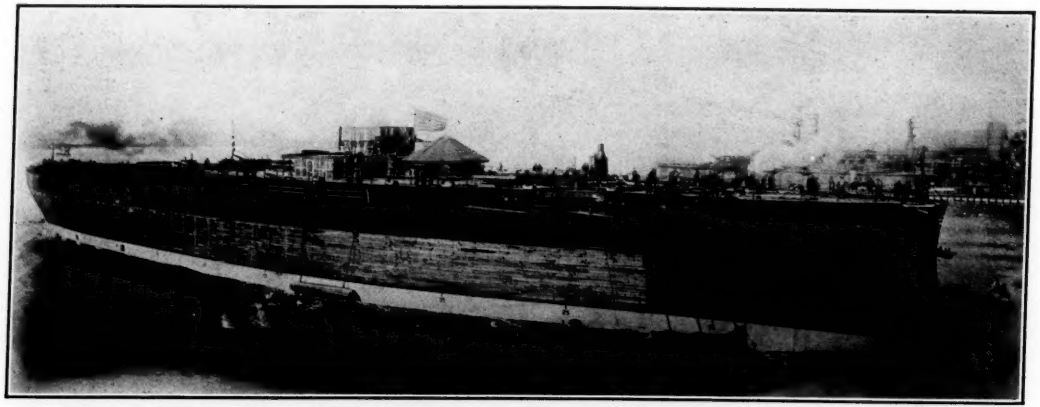
## COMPARATIVE COSTS OF WARSHIPS BUILT IN GOVERNMENT NAVY YARDS AND IN PRIVATE SHIPYARDS.

In the discussion of the Naval Appropriation Bill before Congress on April 21, Mr. William A. Jones, of Virginia, gave some interesting figures regarding the relative costs of naval vessels built in Government navy yards and by contract in private shipbuilding establishments. The figures for the most part were supplied either by the Secretary of the Navy or by the Chief of the Bureau of Construction and Repair, and include, besides the first cost of such vessels, a record of the cost of repairs on the ships since they have been in service. Taking the case of the battleship *Connecticut*, which was built eight years ago in the New York navy yard, it was shown that the *Connecticut* cost $374,000 (£76,600) more than a sister ship, the *Louisiana*, built by contract at the yards of the Newport News Shipbuilding & Dry Dock Company, Newport News, Va., the cost of repairs on the *Connecticut* up to March 1, 1914, amounting to $31,000 (£6,360) more than the cost of repairs on the *Louisiana* up to the same date. Again, in the case of the *Florida*, which was also built at the New York navy yard, the first cost of the ship was $2,269,000 (£465,000) more than that of the *Utah*, a sister ship, built by contract at the New York Shipbuilding Company, Camden, N. J. The cost of repairs on the *Florida* up to March 1, 1914, amounted to $55,000 (£11,250) more than the cost of repairs on the *Utah*. Still another example for comparison was offered by the completion this year of the battleships *New York* and *Texas*. The *New York* was built in the New York navy yard, and cost $1,463,000 (£300,000) more than the *Texas*, a sister ship built by the Newport News Shipbuilding & Dry Dock Company, Newport News, Va. Notwithstanding that the total expenditures on the battleship *New York* up to March 1, 1914, indicate savings of $413,750.50 (£84,700) for the construction of the hull and $245,884.91 (£50,400) for the construction of the machinery as compared with the original estimates for the ship, and that additional amounts above the contract price of the *Texas* were paid by the Government for extra work on the *Texas*, nevertheless the Government-built ship *New York* cost the Government more than a million dollars more than the contract-built ship *Texas*. At present the Navy Department is requesting an appropriation for the construction of two battleships, eight destroyers and three submarines, the cost of which, exclusive of armor and armament, according to estimates of the Navy Department, would be, if built in private yards, about $25,600,000 (£5,250,000), while if the ships were built in navy yards the cost would be about $32,003,000 (£6,565,000), or about 28 per cent. more than the cost under private contract. The significance of these figures led Mr. Jones to oppose very strongly the policy of the Government in placing large contracts for the construction of naval vessels with navy yards, only two of which are at present properly equipped for building large ships, whereas there are now in the United States no less than five large private shipyards which have been developed in recent years very largely for the purpose of undertaking such work, and which have amply proved their ability to turn out the high grade of work demanded by the Navy Department under the most rigid inspection at a much less cost than is possible in navy yards.



U. S. S. "OKLAHOMA" JUST BEFORE LAUNCHING.



U. S. S. "OKLAHOMA" AFTER LAUNCHING.

## *OKLAHOMA.*

The first United States oil-burning battleship, the *Oklahoma*, was launched at the yard of the New York Shipbuilding Co., Camden, N. J., March 23, 1914, and christened by Miss Lorena Jane Cruse, daughter of the governor of Oklahoma.

The following are the particulars of this vessel:

| | |
|---|---|
| Length over all, feet | 583 |
| Length on water line, feet | 575 |
| Beam, feet and inches | 95–3 |
| Draught, feet and inches | 28–6 |
| Normal displacement, tons | 27,500 |
| Full load displacement, tons | 28,400 |
| Fuel capacity (oil only), gallons | 598,400 |
| Equivalent coal capacity, tons | 3,000 |
| Engines, two, 4-cylinder triple, 35-inch × 55-inch × 78-inch, 48-inch stroke. | |
| Total I.H.P | 24,800 |
| Boilers, twelve B. & W.; heating surface, square feet | 48,000 |
| Weight of machinery, tons | 1,900 |

### BATTERY.

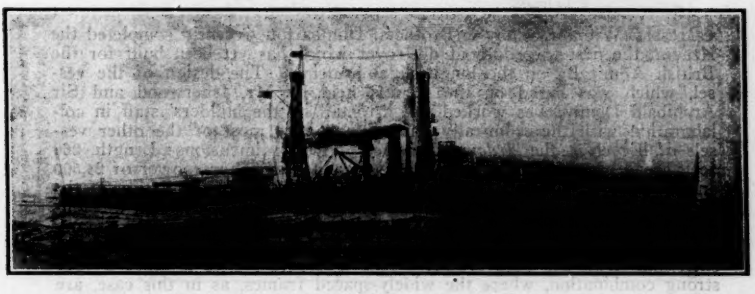
Ten 14-inch guns in four turrets; all on center line as shown in photo. The end turrets have three guns each; the inner turrets two each.

Secondary battery, twenty-one 5-inch guns, four 3-pounders, two 1-pounders, and four submerged torpedo tubes.

### ARMOR.

The main belt is 13½ inches thick, 17½ feet wide; extends 8½ feet below water line. The triple turrets have 18-inch armor, the double turrets 16-inch. There is a 3-inch protected deck. There is one smoke pipe, protected at base with 13½-inch armor.

The accompanying photos show the vessel at time of launching, while the photo drawing shows how vessel will appear when completed.



INT. MARINE ENGINEERING.

U. S. S. "OKLAHOMA."

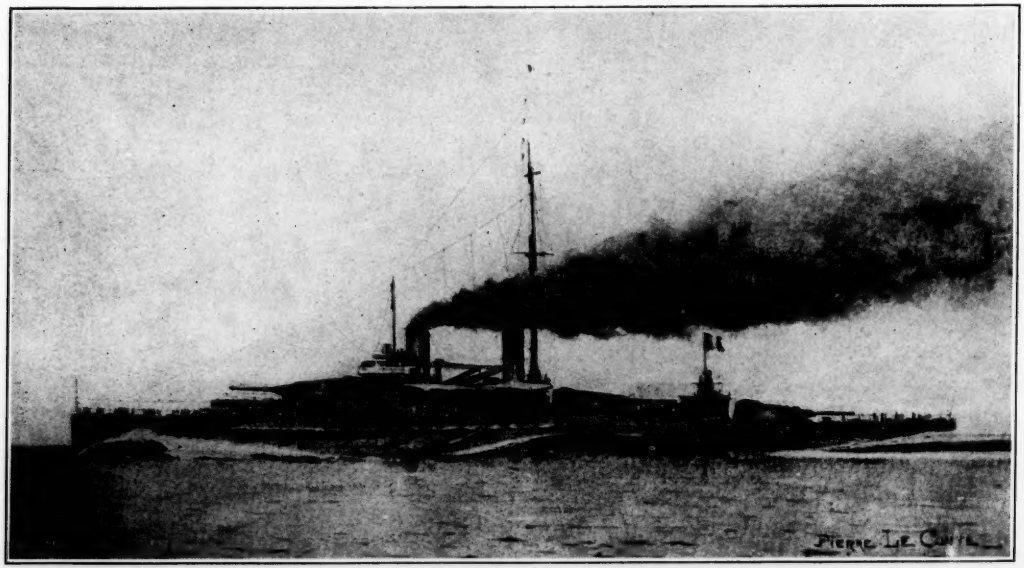
## BRITISH.

### THE NEW BRITISH BATTLESHIPS.

The Admiralty have now asked for tenders for the two battleships of this year's program which are to be built by contract, and the expectation is that it will be possible to give out the contracts early in June. The vessels are to be in principle "Royal Sovereigns." They will be, that is to say, of a length between perpendiculars of about 580 feet, and, with steam turbines, will have a speed of at least 21 knots. Their main armament will consist of eight 15-inch guns, and their anti-torpedo-boat armament of sixteen 6-inch guns. Of the two other large ships of the year one is to be of the *Queen Elizabeth* type and one of the *Royal Sovereign* type. They are to be built in the dockyards, and are not to be laid down until late in the year. Although the fact has not been stated officially, it is understood that invitations to tender for the contract ships have been sent to nine firms. Since the Thames Ironworks Company ceased operations, only eight private concerns have been on the Admiralty list of armored shipbuilders. These are the Fairfield Company, of Govan; John Brown & Co., of Clydebank; William Beardmore & Co., of Dalmuir; Scott's Company, of Greenock; Armstrong, Whitworth & Co., of Newcastle; Palmer's Company, of Jarrow; Vickers, of Barrow; and Cammell Laird & Co., of Birkenhead. Swan, Hunter & Wigham Richardson, of Wallsend, although they are not believed allied with any armor plate makers, have, we believe, now been added to the list. They are, of course, already Admiralty contractors in a large way, and having built a ship like the *Mauretania* there is no reason why they should not build one like the *Royal Sovereign*. They have at present in hand for the Royal Navy a light cruiser and a destroyer. The time specified for the construction of the new battleships is 27 months, and with stocks of work declining everywhere there ought to be no difficulty in adhering to it. Practically all the firms tendering have Dreadnought-berths vacant, so that very little time should elapse between the placing of the contracts and the laying down of the ships, no matter who gets the work.

### AN "ISHERWOOD" DESTROYER.

Messrs. William Denny & Brothers, Dumbarton, recently completed the *Ardent*, the first torpedo-boat destroyer which has yet been built for the British Admiralty on the longitudinal principle. The design of the vessel, which was based on the patents held by Mr. Isherwood and Sir Archibald Denny, was worked out in detail by the builders' staff in collaboration with the Admiralty authorities. Like most of the other vessels of her class, the *Ardent* is of the following dimensions: Length, 260 feet, breadth, 28 feet, and depth, 9.2 feet. She has a horsepower of 24,500 and a speed of over 29 knots. The principal feature in the design of the vessel is that the frames are carried out to the shell direct, and the longitudinals are cut at each frame and bulkhead, being fished at the frames by short pieces of angle passed through holes inside of the line of the shell angles on the deep frames. This is considered to produce a very strong combination, where the widely-spaced frames, as in this case, are necessarily of limited depth. At the bulkheads the longitudinals are connected by knees on each side, thus dispensing with a great deal of collar work usually associated with destroyers.



FRENCH BATTLESHIP "BÉARN."

## GERMAN.

### OIL FUEL IN THE GERMAN NAVY.

From "Suddeutsche Zeitung," Stuttgart, April 18, 1914.

If England has gone over to oil fuel on their warships much faster than, for instance, Germany, there are good reasons for it. It is an open secret that the full manning of ships in the English Navy causes much racking of the brain. As the serving of oil boilers requires only one-third of the men necessary for coal firing, it can easily be grasped that England desires oil fuel for her navy. Exclusive oil fuel was introduced the first time in 1905, on the destroyers laid down in that year. The first large battleships with exclusive oil were the battleships *Nevada* and *Oklahoma* of the United States. The question arises, shall we follow the British example and gradually introduce the oil fuel.

The great advantages of oil fuel cannot be denied, but the fact must be conceded that a mixed fuel system, as is used in the German Navy, suits our conditions better. We do not desire to be deprived of the advantages of oil fuel, but we also do not care to burden ourselves with the disadvantages. On those German ships, therefore, building to 1911, arrangements were made both for oil and coal fuel. A part of the boilers is installed for both oil and coal purposes, for instance, on the *Graudenz*. On account of the high price of oil, and partly on account of the difficulties connected with procuring the same, our ships are using coal as fuel in normal trips, and resort to oil as fuel only then when high speed is to be attained.

The main advantage of oil fuel lies in the attainment of high speed and increased radius of action, thereby heightening the strategical value of the ships. Experience has proved that one kilo fuel oil renders the same work as 1¼ coal, the proportions of weights of oil and coal as far as their fuel value is concerned is, therefore, like 14:9. Besides, oil takes up 40 per cent. less space than coal, so that a much larger quantity of oil can be taken than coal.

The advantages are confronted, however, by very important disadvantages. In adopting the oil as fuel exclusively we would be dependent on the United States and Roumania. Both of these countries export this article in quantities worth mentioning. The procuring of oil would become still more difficult for our Navy if internal-combustion motors are to replace the steam engine before long, which is not at all improbable. Then the oil consumption of Germany shipping would rise to 1.1 million tons (1 ton = 20 Zentner). The whole production of fuel oil in Germany is only at the present time 521,000 tons. We would then be even more dependent on foreign countries for the supply of oil. It would also be necessary to have in time of peace large storages of oil on hand. The storage of such large quantities of oil are an immense danger from a military standpoint, as these storages can be destroyed by explosives thrown from air craft and thus destroy the usefulness of a fleet which is exclusively fitted up for oil fuel. It will therefore be necessary to continue the double system of oil supply in Germany as it exists at present.

# MERCHANT SHIPS.

## THE NEW WHITE STAR LINER *BRITANNIC.*

With the launch of the *Britannic* February 26 another mammoth ship enters the water, and although not so big as the *Imperator,* she will rank as one of the biggest in the world.

The following are the leading particulars of the *Britannic:*

| | |
|---|---|
| Length, overall, feet | about 900 |
| Breadth, extreme, feet | about 94 |
| Depth, molded, feet and inches | 64–3 |
| Total height from keel to navigating bridge, feet and inches | 104–6 |
| Gross tonnage, tons | about 50,000 |
| Load draught, feet and inches | 34–7 |
| Displacement at load draught, tons | over 53,000 |
| I.H.P. of reciprocating engines | 32,000 |
| S.H.P. of turbine | 18,000 |
| Passenger accommodation | over 2,500 |
| Crew accommodation | over 950 |

Each successive Atlantic liner launched at the Belfast works of Messrs. Harland & Wolff, Ltd., has marked a distinct step, not only in size, but in strength and in those provisions necessary to ensure comfort to the passengers and the greatest degree of safety which is possible considering the elemental and other forces of nature. Indeed, it may be said that each vessel typified the highest achievement of her day in the practice of shipbuilding and marine engineering. The *Britannic,* the latest of the ships, is in every respect an example of this periodic standard of progress, but as she is only of slightly greater tonnage than the *Olympic,* being about the same length but rather more beam, it would seem that the owners—than whom no one has a better conception of the requirements of the Atlantic passenger trade—have come to the conclusion that for the present the traveling public are completely satisfied with the *Olympic* type of ship. Nor is any attempt to be made to increase the speed, the schedule time will be maintained with precision by machinery power sufficient to enable the average service speed of 21 knots to be maintained on any voyage irrespective of the most adverse weather which may be met with either for a part or for the whole of the time the vessel is on the Atlantic.

Hitherto the advance in size has been considerable in the case of each of the successive large ships. Thus, in the *Oceanic,* completed in 1899, the length was 685 feet, and the displacement 31,600 tons at 35 feet 7 inches draught. The *Cedric,* built in 1903, had a length slightly less, but, owing to her greater beam, an increased displacement—37,900 tons, on a draught of 36 feet 8 inches. The *Baltic,* of the following year, had a length of 708 feet and a displacement of 40,700 tons at a draught of 37 feet 3 inches. The *Adriatic,* completed in 1906, was the next notable ship, and in her case the length was over 725 feet and the displacement 40,800 tons. The *Olympic,* completed in 1911, marked a still further increase in length, to 882 feet 9 inches, with a displacement of 50,000 tons at 34 feet 6 inches

draught. Now comes the *Britannic*, with slightly increased beam, the result being that the displacement is over 53,000 tons at 34 feet 7 inches draught. These successive increases in size, it will be seen, have been confined within reasonable limits, and the aim underlying the design has been at all times not only to ensure a dividend-earning ship, but to make certain that the vessels would stand up to the greatest of Atlantic gales, and in other respects would conduce to the comfort of passengers on board.

STRUCTURAL FEATURES.

Messrs. Harland & Wolff have probably a greater experience than any single shipbuilding firm in the design of the structural details of these modern behemoths, and the *Britannic* embodies every lesson of accumulative experience. Her double bottom, which is over 5 feet deep, increased to over 6 feet throughout the machinery space, has on each side of the center line six great longitudinal girders which, with transverse plating, divide the double bottom into a considerable number of cellular spaces. As in the *Olympic*, since the recent alterations, this double system of construction is carried up the sides of the ship to a considerable distance above the load water line, and here again the space between the two skins is divided, not only by vertical partition walls, but by heavy horizontal watertight plating; so that here, also, there is a great number of small compartments. The advantage claimed for this system of construction as compared with side coal bunkers serving the same end—the providing of a double skin—is that there is never during the voyage an open door in the inner wall of the double sides, as with coal bunkers. The framing of the ship throughout is exceptionally heavy, and extends from the bottom of the ship to the shelter deck, a height of 66 feet. Moreover, hydraulic riveting has been introduced to a larger extent than formerly, with, in some places, four lines of rivets. This applies particularly to the top shell plating, which, as in all ships, is subjected to the greatest stress when the ship is borne forward and aft on the crest of waves or when she is carried only amidships, the bow and stern being over the wave troughs. Again, there extends, right fore and aft, at the level of every deck, four lines of heavy girders, and at frequent intervals, extending from the bottom of the ship right to the shelter deck, stanchions and heavy columns, thus ensuring great longitudinal and vertical stiffening to the whole structure. The deck beams assist to the same end transversely, and have been made especially strong. Even where there are openings for the boilers and machinery an equally great degree of structural strength is imparted by the use of girders in the longitudinal and transverse lines, supported again on heavy columns. These facts show that in respect of strength the ship agrees with the progressive practice of Messrs. Harland & Wolff.

There are 16 transverse bulkheads, five of which extend to a height of over 40 feet above the deepest load line, while all the others are carried to a height of over 21 feet above the water line. The bulkheads are of very heavy construction, the aim being not to economize weight, but rather to ensure the maximum of strength by a careful distribution of it where it is most required. The stern framing and the brackets for carrying the three propellers are of very massive construction, a fact which is established by the great weight of the forgings and castings, namely, over 300 tons. The rudder, too, weighs over 102 tons, and is operated by Messrs. Harland & Wolff's steering gear, which has a toothed quadrant, operated to port or starboard by means of two powerful three-cylinder engines, as large almost as those for the propulsion of a fair-sized cargo boat. The engines are in duplicate, so that there is no possibility of the breakdown of one preventing the steering of the ship. Provision is also made for taking up any shock due to a green sea striking the stern of the ship, as this might otherwise affect the steering machinery.

PASSENGER ACCOMMODATION.

There are nine decks in the ship, on six of which accommodation is provided for about 2,600 passengers, including 790 in the first-class and 830 in the second-class.. A feature of the accommodation is that a large number of the first-class state rooms are single-berth rooms. In practically every case, whether single or double-berth room, there is in connection with each a bathroom and closet or a shower bath and closet. There are a large number of special suite rooms on the bridge and shelter decks. And there are also two special suites, these including a sitting room, two bedrooms, servants' rooms, bathrooms, &c., and an isolated veranda about 25 feet long on the starboard side, while the suite on the port side has an isolated deck promenade about 50 feet long. Extending through the first-class passenger decks there are two large companionways. In that forward there are three electric elevators extending right up to the boat deck, while further aft, but independent of the companionways, is another first-class elevator.

As regards the public rooms there are on the bridge deck a large gymnasium and a children's play room. On the promenade deck, the next level, a feature is that the plating of the ship is carried up the side and has large windows to enable either a winter garden to be arranged or to provide for passengers a sheltered promenade with a complete view of the sea. The public rooms on this level include a reading and writing room and a first-class lounge, as well as a first-class smoking-room with veranda café and palm court. On the bridge deck aft there is an *a la carte* restaurant (the full width of the ship) with pantry, &c. The next level, the shelter deck is, so far as first-class passengers are concerned, taken up entirely with state rooms, while on the deck below is the dining saloon and the reception room in connection with it; the two together occupy nearly 200 feet of the length of the ship. On a still lower deck there are Turkish and electric baths, racquet court, and a swimming bath.

The second-class accommodation is little inferior to that of the first-class and is equally adequate. It is arranged immediately abaft the first-class quarters, with a companionway and electric elevator extending right up to the boat deck, the dining saloon being on the saloon deck, the second-class library and reading room and second-class gymnasium on the shelter deck, where also there is an extensive promenade, and a smoking room and large space for promenade on the bridge deck. The third-class public rooms are in the stern of the ship, excepting the dining saloon, which occupies the space of two watertight compartments on the middle deck amidships, the length of the two rooms combined being 100 feet, and the full width of the ship. The companionways for the third-class are greatly increased in number.

New features, too, have been introduced in connection with sanitation. Probably for the first time in shipbuilding an effort has been made to depart from the methods which have been handed down from the beginning of shipbuilding. Instead of an enormous number of soil discharge pipes from lavatories, &c., on each side of the ship, the arrangement simulates, in each compartment, the conditions applicable to the best town planning system, with the assistance of electric pumps. The hot-water system, too, has had special consideration, and the system adopted is such that immediately a tap is turned on hot water is obtainable, instead of a large quantity of cold water preceding the flow of the warm water. Thus there is great saving in the fresh-water supply, an important desideratum on board ship where any shortage must be made up by distilling. In respect of ventilation, too, experience has been very carefully collated, so as to arrive at the highest efficiency. The dominant idea is to extract the foul air from the interior and to pass fresh air—which may be heated in

the winter time—through trunks along the side of the ship on each deck, having louvres at various points; there are as many as 1,500 such louvres in the vessel. In addition, every first-class room has an electric fan, while the public rooms are also ventilated and heated by electric fans. In the principal entrances there are large electric radiators having a capacity of from 3,500 to 6,000 watts. Electricity, indeed, is very extensively utilized throughout the ship, the power station being as great as that in many provincial towns.

### ELECTRICITY SUPPLY.

The total generating power in the ship is equal to 1,660 kw., and perhaps a clearer idea of this capacity is afforded by the fact that the current, if used entirely for lighting, would be capable of running over 150,000 lamps of 10 c.p. A large proportion of the electricity generated is utilized for power purposes. There are electric ventilating fans, elevators for passengers, mails, baggage, &c., electric radiators, electric cranes for manipulating the cargo, two electric whistle controls, and electricity is employed for the telephone system throughout the ship, for the loud-speaking telephones between the bridge and the various working departments of the ship, and for electric baths, down even to electric machines for clothes pressing and for heating the tongs in state rooms and in the ladies' hairdressing department, the last-mentioned a luxury even in Atlantic ships. There is, further, the Marconi installation, the wires of which are carried on the masts which rise to a height of over 200 feet above the load line. The power of the installation is sufficient to transmit messages over 2,000 miles, so that the vessel will be in communication with either Britain or America, and for a considerable time in mid-ocean with both countries.

As in some preceding ships built by Messrs. Harland & Wolff, there is what is termed an emergency electric-power station on one of the decks far above the water line. In this there is not only a very extensive series of electric storage batteries which are always charged, but two powerful electric generating machines, which can be used not only for charging the accumulators when current is being taken from them, but for running direct what is termed the emergency circuit, including the Marconi apparatus, the electric whistle controls, the electric motors working the boat hoisting gear, lighting the more important parts of the ship, and doing such other work, so that when the main power station is not of use the necessary electric work can be continued. This set, too, will suffice for providing current for all electric operations and lighting necessary when the vessel is in port.

An interesting feature in the ship is the complete control which the captain exercises from the navigating bridge, which is 63 feet above the water line and 100 feet above the keel. From there orders can be given by telephone or telegraph to every working quarter of the ship, and instruments are provided to demonstrate that most of the important orders have been carried out. Thus, there is an indicator to show the working of the main engines, the operation of the steering machinery, and the actual position of the bulkhead doors, the arrangement in this case being such that progress in opening or closing the door is automatically shown. Pneumatic tubes are provided for the receipt and despatch of messages to the Marconi room. The angle of the rudder is electrically recorded and the depth of the ocean is sounded by electric machinery, while electric submarine bells indicate the proximity of lightships, &c. The lookout men have telephonic communication with the bridge, and the steering is done through telemotor gearing direct. Telegraphs indicate the necessary instructions to the men in charge of the anchors and capstans. There are five anchors, the largest being 16 tons, and powerful steam windlasses and warping gear are provided.

### LIFEBOAT EQUIPMENT.

Forty-eight of the largest size of lifeboat yet made are being fitted, and two of these have powerful engines. These boats, instead of extending right along the boat deck, are arranged in four separate groups with abundant room for marshaling passengers, &c. The system of davits used differs from that in any other preceding ship. There are two davits on each side of the deck, where the boats are placed. These do not slew, the space apart being sufficient to pass the boats through. They are of lattice-girder construction with a swan-necked top turned towards each other in each pair. They more resemble shear legs in their action than davits or cranes, being pivoted at their base and moving from the vertical position to a considerable angle inboard or to a considerable angle outboard. Indeed, the angle is so great that the davits command one-half of the deck of the ship, while when outboard they will enable the boats to be lowered vertically into the sea, even if the vessel has considerably heeled over. The arrangement is such, too, that the boats may be traversed across the deck, so that all the boats may be lowered on the one side of the ship at the will of the captain. The davits are inclined inboard or outboard by means of powerful screw gear. From the stem of each davit there extends inboard a built-up stay, the inner end of which is secured to a nut mounted on a worm shaft on the deck. By electric motor the two worms for each pair of davits are rotated, and as the worm screw rotates the nut travels along it, and, acting through the stay, pushes the two davits from the vertical to a considerable angle outboard. The reverse travel of the two nuts on the worm pulls the davit from the vertical to a considerable angle inboard in order to lift the boats from their chocks on the decks. The height and outreach of the davit enables the boats to be mounted one over the other in tiers, and also facilitates the placing of several tiers in the width of the ship. Moreover, the davits can be kept so far inboard as to give a wide passage at either side for promenading and for marshaling the passengers in case of emergency. Limit switches are provided, so that in the event of any accident to, or temporary aberration on the part of the man manipulating the gear, the motion of the davits or boats will be arrested before any damage will take place, thus making the gear practically mistake proof.

Powerful electric lamps are provided at the top of each davit, which not only assist in illuminating the surrounding space, but greatly facilitate the operation of the gear at night. Another important feature in the invention is the arrangement by which boats can be lowered on an even keel even in the event of the ship being down by the head or the stern. The two falls are wound on separate drums mounted on the same shaft, and normally rotate together when a boat is being raised or lowered, but means are provided for rotating one of the drums relatively to the other, and thus wind up or pay out its fall independently of the other fall. A further advantage of the Harland & Wolff davit is the fact that the boats can be all open lifeboats of good type.

### PROPELLING MACHINERY.

The three screw propellers are operated by combination machinery consisting of two sets of four-cylinder triple-expansion reciprocating engines operating the wing shafts, and a low-pressure turbine of the Parsons type driving the center shaft. The steam, after passing through the successive cylinders of the reciprocating engine, flows normally into the turbine, and although the pressure at entrance is only about 10 pounds absolute, as compared with about 200 pounds at the reciprocating engines, the exhaust turbine enables the whole of the energy in the exhaust steam to be utilized before it passes to the condenser. Thus the turbine, notwithstanding the low initial steam pressure, develops about 18,000 H.P.,

which is slightly more than the power developed by each of the two reciprocating engines. The result is very much higher economy than is possible with reciprocating engines only. Messrs. Harland & Wolff have made a specialty of the combination type of machinery; indeed, no firm has done more in the adoption of this system than the Belfast firm, who have completed, or have in hand, such machinery for 450,000 H.P. The consequence is that they are able so to proportion the engines as to get the best results. The *Britannic* has a most powerful set of such combination machinery, and the turbine, which has been made entirely at the Belfast works, is one of the largest marine exhaust turbines yet made, the weight of the turbine complete being close upon 500 tons. The turbine exhausts its steam into two condensers, but provision is made by means of a special valve so that the exhaust steam from the reciprocating engines may pass direct to the condenser instead of through the turbine to the condenser. This confers the advantage that no reversing or astern turbine is necessary on the center shaft. The astern driving of the ship, and therefore all the maneuvering in port, is done by the reciprocating engines on the wing shafts. The "change over" from driving ahead with all three propellers to operating the wing shafts, as in the ordinary twin-screw ship with reciprocating engines, is effected almost instantaneously. In maneuvering, the ship corresponds exactly, so far as the manipulation of the machinery is concerned, to a twin-screw vessel.

### BOILERS AND FURNACES.

There are in the ship 29 boilers, of which 5 are single-ended, the remainder being double-ended. These are distributed in six boiler rooms separated by watertight bulkheads and by coal bunkers athwart the ship between the two inner skins and up to a level far above the load water line. Thus there are practically three division walls between each boiler room, with a very considerable intervening enclosed space. Stone's underline ash expellers are provided in each boiler room, also ash hoists supplied by Railton, Campbell & Crawford. For forced ventilation of the boiler rooms twelve Sirocco fans have been supplied by Davidson & Co., of Belfast, fitted with Allen motors.

In all there are 159 furnaces in the ship. There are four funnels, which rise to a height of about 180 feet from the bottom of the ship. There is a complete system of auxiliary machinery throughout the ship and very complete arrangements have been made in connection with the pumping of the compartments. It is possible independently to pump any compartment, and the valves in connection with the pumping can all be operated from one of the decks far above the water line.

The main air pumps have been supplied by G. and J. Weir, Ltd., also fresh water and oil pumps. The circulating pumps in connection with the condensing plant have been constructed by Harland & Wolff, but the others principally by Messrs. Weir, Ltd.

### LAUNCHING.

The vessel was held on the ways by hydraulic triggers, only requiring to be released by the opening of a valve to let the structure glide into the water. The launching weight was 24,800 tons, and the launching gear was situated about 40 feet from the port bow of the ship on the ground. Mr. Payne was stationed at the gear with his eye on the gage, and at a signal from him, by a simple turn of the controlling lever, the valve was opened which released the triggers and the vessel moved, imperceptibly at first, slowly gaining momentum as she took advantage of the declivity of the ways. The total time of the launch was officially returned as 81 seconds, and the maximum speed attained on the ways was 9½ knots. A depth of 50 feet of water was provided in the river, and this proved more than amply sufficient. The total depth of the stern in the plunge

into the water was 31 feet, while the depth of the stem plunge was 17 feet.

The arrangements for checking the vessel were of an elaborate nature, owing to the somewhat narrow waterway. In the bed of the river there were placed three heavy anchors on each side of the ship, each anchor being connected by a steel wire hawser to eyeplates riveted to the shell plating. There were also placed in the river two piles of cable drags each weighing over 80 tons, connected in a similar manner with a steel-wire hawser. These were all arranged so that when the vessel was clear of the end of the slip the drags and anchors acted simultaneously in bringing the ship to a standstill. The 10-inch anchorage rope (with a guaranteed breaking strain of not less than 305 tons), 5-inch mooring ropes, and other ropes used in connection with the launch were supplied by Messrs. Bullivant & Co., Ltd. As a further precaution, the ship's own bow anchors were stowed in the hawser pipes ready for letting go in case of emergency. So effectual were these methods that from the time the triggers were released until the vessel was stationary in the water, only two minutes elapsed.—"Shipbuilding and Shipping Record."

NOTE.—A full description of the details of construction and parts of machinery is given in "Engineering," February 27, 1914. This is fully illustrated with illustrations and drawings showing the principal parts of engines, boilers and auxiliaries. This description is probably more complete than any ever published on a merchant vessel. As "Engineering" is available to most of the readers the full description is not reproduced.

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## THE MOTOR SHIP *SEBASTIAN*.

This Diesel engined vessel of 1,600 I.H.P. and her machinery are fully described in "The Engineer," March 6, 1904. The engine by A. B. Diesel's Motorer, Stockholm, presents various novel features which are fully commented upon. This description will be found of special value to those interested in the latest practical development of marine Diesel-engine installations.

# OBITUARY.

## GEORGE WESTINGHOUSE.

By Walter M. McFarland.

The end of the earthly career of one of the giants of engineering and industry came on March 12, 1914, when George Westinghouse passed away, after a life filled with brilliant achievement and benefaction to the race in a degree which has rarely been equalled and probably never surpassed. Although not an old man as age is counted today, he had been famous so long that the few remaining old men who actually saw the beginning of his fame had forgotten the date, and the younger men always thought of him as one of the great institutions in the engineering world. The scope of his genius was so great that he touched many sides of engineering and industry, and in each has left monuments which will perpetuate his memory as long as there is a student of the history of engineering.

When the complete biography of Mr. Westinghouse is written by some literary artist, as it should be, the story will be one of surpassing interest, not only from the narration of the deeds actually accomplished, nearly all of which tended to the amelioration of labor and the increase of safety, but in the portrayal of a character which has always excited the admiration of those who knew it best, and was specially marked by courage, energy, tenacity of purpose, vision, executive ability and soundness of judgment. If there are any who think that the invention, development and exploitation of mechanical devices is a prosaic career, the life of Mr. Westinghouse will prove to them that in his case at least it had all the elements of romance, and often of intensely dramatic situations.

He was undoubtedly a great genius, a great inventor, but in addition he had wonderful judgment as to what was practical and useful, and the executive and administrative ability to plan, organize and conduct the works necessary to utilize completely the inventions he had made.

It is a common but erroneous notion that inventions are happy inspirations which come without effort to the lucky inventor. The history of every one of Mr. Westinghouse's inventions is the complete denial of this, for he was led to each of his discoveries by familiarity with the field of its application and with the need of some device to accomplish the result which his invention secured. Thus, in the case of the air brake, an earlier invention for replacing derailed cars brought him in touch with railroading, and he observed that the facilities for braking were entirely inadequate. The characteristics of his mind which made him a genius always led him in such a case to endeavor to devise a remedy. His first idea was a brake to be operated by steam, and the cylinder and rigging under the car were devised with this idea, substantially as applied later. He realized the drawbacks to the use of steam, and, while studying the problem, read of the use of compressed air in the piercing of the Mont Cenis tunnel for driving the rock drills. He saw at once that this gave him the suitable agent for transmitting his power, and the air brake was born in 1867. While an immense advance on anything which had preceded, it was still open to improvement; and some years afterwards his invention of what is known as the "triple valve" made the air brake essentially what it is today, although improvements have continued down to the present time. It is hardly too much to say that the invention of the air brake has made modern high-speed railroading possible, and everyone who rides in a fast train is a debtor to George Westinghouse for the safety and the ease of mind with which he journeys.

The invention of the air brake having made higher speeds possible, improved switches and signals became necessary, and

Mr. Westinghouse became interested in this line of work, which resulted in the Union Switch and Signal Company, the second of his industrial establishments, the air brake being the first. The product of this company is probably most familiar to travelers by the overhead bridges carrying semaphores, which, supplementing the air brake, have enormously increased the safety of travel as well as the capacity of a line for passing a large number of trains.

In the operation of the signals electricity was employed, and the familiarity thereby gained led Mr. Westinghouse to what many consider the greatest work of his life, the introduction and successful operation of alternating-current electricity. The form of electricity in general use at this time (the early '80s of the last century) was direct current, but, with the vision of genius, he realized, as apparently no one else then did, that the possibilities of electricity for lighting and power could never be fully utilized with direct current, owing to its serious limitations. He began, accordingly, to develop the use of alternating current for arc lighting, which at that time seemed its only field. He saw that two vital things were necessary to the complete utilization of alternating current, one a device for raising and lowering the voltage, and the other a simple motor. With the former, domestic lighting would become possible; and, in fact, the use of current anywhere, as the voltage could be adjusted to suit. With the simple motor the application of alternating current for power purposes would be easy. The former became possible through the invention of the transformer by Gaulard and Gibbs, and the latter through the invention of the induction motor by Nikola Tesla. The Westinghouse Electric and Manufacturing Company, which became the largest of all his enterprises, was started in a modest way for the arc lighting. With the transformer and induction motor it grew in size and was arranged to include also the manufacture of all forms of direct-current apparatus. It is important to mention, as one of the romantic and dramatic episodes in Mr. Westinghouse's career, that the introduction of alternating

current, far from being welcomed, as might be supposed today, was bitterly opposed. The opposition was, indeed, so intense that only a genius with the tremendous personality and tenacity of Mr. Westinghouse could ever have made any headway at the time. An effort was made in some States to have its use declared criminal, and it is interesting to know that, when electrocution was adopted in New York, the first dynamo used was a Westinghouse alternator which had been purchased in a round-about and clandestine way so that its destination was not known. After it had been installed, the opponents of alternating current used its selection as an additional argument to prove the specially deadly effect of alternating current.

Courage has been mentioned as a marked characteristic of Mr. Westinghouse, and many of the earlier contracts taken by the Electric Company demonstrated it in a marked way. He took the contract for lighting the World's Fair in Chicago in 1893, but was then enjoined for patent reasons from using the form of lamp which his factory had been making. Nothing daunted by this, he invented what was known as the "stopper" lamp, which enabled him to fill the contract. Those who were at Chicago will remember the wonderful effect given to this first of the Great Fairs to be illuminated by electricity, by these "stopper" lamps with current furnished by Westinghouse alternators.

Perhaps an even greater illustration of his courage was the building of the first generators at Niagara Falls. The machines were not designed by the Electric Company, and at a later date much simpler and more satisfactory ones became the rule, but he had the courage to undertake the manufacture of these absolutely novel and, for the date, enormous machines, and made them a great success. In the last decade of the last century came the great machines which furnish current for the elevated railways and subway of New York, and in the first decade of this century came the alternating-current railway motor and locomotives on the New Haven Railway and the enormously powerful direct-current locomotives of the

Pennsylvania Tunnel under New York City. The wonderful work which Mr. Westinghouse did in electrical development was not in the way of personal invention, but in the marvelous foresight which could plan for ultimate success and knew how to adapt the various elements to bring that success about.

The Westinghouse Machine Company afforded another field for his invention and administrative ability along mechanical lines, which was where he was most thoroughly at home. Organized to manufacture the Westinghouse steam engine, the Machine Company also took up the development of the gas engine, where, as usual, he was in the lead. When Sir Charles Parsons had brought the steam turbine into commercial possibility, although in small sizes, Mr. Westinghouse, as usual, foresaw its wonderful possibilities, and obtained the rights for America. With him, however, the possession of any new device was a challenge to its improvement. This at first took the form of building larger sizes in a tentative way, and, owing to his efforts the Parsons turbine, as built in his shops, was for several years steadily in advance of the sizes built anywhere else. As the power increased, he saw that a great improvement could be made in the space occupied, and this led to his invention of what is known as the "double-flow" turbine, which greatly reduced the space occupied, and by the combination of impulse blades to receive the high-pressure steam with reaction blading to utilize its expansion, set the type which all forms of turbines are taking in their design today.

The working out of the double-flow turbine occurred within the writer's personal knowledge, and it illustrates in such a marked way some of Mr. Westinghouse's striking characteristics that the story is well worth telling. His method in getting out new work of this sort was to sketch and talk it over with his engineers and have them lay out the drawings for his inspection. After he was satisfied with the drawings the machine was constructed, to be tested and modified in accordance with his ideas for improvement. Everyone of the engineers of the Machine Company tried to dissuade him from working on the

double-flow turbine, asserting that it was impossible for this form ever to give the economy of the older form known as the single-flow turbine. Almost any other man in the face of such united opposition from engineers of great ability would have been inclined to abandon the effort, but not Mr. Westinghouse. It was not the first time that he had stood alone in his opinions only to be proved right in the end. He had reasoned the matter out thoroughly, was satisfied that he was right, and believed that his judgment was better than that of his engineers. The early tests of the first one constructed seemed to prove that the engineers were right, for the economy was undoubtedly lower than that of the single-flow turbine. Not discouraged, he made one modification after another, with the result that finally the economy was at least as good, with the great advantage that it made possible very large powers which were almost impracticable in the older form.

One of the latest of his inventions was in connection with the reduction gear brought out originally for securing high economy in driving the propellers of steam vessels, but finding a very useful field also in the driving of direct-current dynamos by turbines where the speed conditions are much the same as on board ship. The reduction gear was the joint invention of the late Admiral Melville and Mr. J. H. Macalpin. Mr. Westinghouse was a great personal friend of Admiral Melville, and became interested in the gear at first on this account, but subsequently was convinced of its great merit. He financed the construction and testing of the first one, and it is interesting to note, as an illustration of what was characteristic with him, that this experimental example of the reduction gear was for the transmission of 7,000 horsepower. As showing his fairness in dealing with people, it may be mentioned that he did not negotiate the purchase of an interest in the gear until after, through his expenditure and the resources of his great shops, its success had been demonstrated, thus placing the inventors in a very much stronger position than if the negotiations had been conducted before the test. While this test showed that the gear was manifestly successful, Mr. West-

inghouse saw a way of improving it, and invented the hydraulic suspension which is now such an essential feature.

It would be impossible in a brief article like the present to give anything like an adequate account of all of this wonderful man's inventions and enterprises, but we must not neglect to refer to his work in connection with the distribution of natural gas. At first, in the various gas fields it seemed only to be necessary to drill a fresh well when an old one gave out, and the distances for transmitting the gas were relatively short and the natural pressure high. After a time, however, nearby fields became exhausted, and it was a question whether the gas could be transmitted over long distances. Most people believed that it was impossible, but Mr. Westinghouse took hold of the problem and devised methods for pumping it, with the result that today Pittsburgh gets most of its gas from wells more than a hundred miles distant.

The world generally knows of the great manufacturing companies which bear Mr. Westinghouse's name, but it is not so well known that he was also at the head of commercial enterprises on almost as large a scale. The Philadelphia Company in Pittsburgh was organized for the distribution of natural gas; and in the early days of the alternating current, in order to demonstrate its economy and convenience, he organized or bought and developed companies in Pittsburgh, New York and Baltimore. When these had become successful, and when the growth of his manufacturing enterprises required more of his attention, these were sold and became ordinary commercial enterprises, entirely independent of the Westinghouse interests.

When it is realized that he was at the head of so many large enterprises in this country and abroad, it is natural to ask how it was possible for one man, however great a genius, to manage all these enterprises and still have time to engage in the minute details which are inseparable from invention. The fact that all this has gone on with such great success for so many years is the most striking proof of his marvelous ability as an executive and administrator. His plan was to surround himself with men of high ability as his lieutenants, to whom he

left all the detail work of administration. He kept in touch with the general work of the companies, but was specially interested in new and original work. Daily and monthly reports were sent to him, from which he had an accurate knowledge of the conduct of the business, but he had the rare gift of being great enough to avoid being overwhelmed with details.

In 1907 the Electric Company, although wonderfully successful, became embarrassed for ready cash, owing to the investment of a large part of its surplus in other interests, largely the foreign companies. This led to a friendly receivership, in order that the company's affairs might be put in better shape. There was much cheap talk to the effect that Mr. Westinghouse, with all his genius and wonderful ability as an executive and administrator, was not a financier. It is always easy to criticise when things are somewhat out of joint, although the critics would not be able to lift a finger towards improvement. The financiers took hold of the problem, and after much cogitation devised a scheme which was solely in the interest of creditors and would have seriously hampered the operation of the Company. When it became evident that this scheme would not be accepted, another was brought out under the auspices of what was known as the Merchandise Creditors' Committee. The details of this scheme were such that those usually recognized as financiers pronounced it impossible, but nevertheless it went through, and was the means of putting the Electric Company again on its feet. It transpired afterwards that this successful scheme was originated and the details worked out by Mr. Westinghouse.

Thus far the story of George Westinghouse has been about the genius and inventor, the executive, the almost superman, but when we are thinking of such a personality we like to get a touch of the human side and find what such a remarkable man was in his daily life and contact with his fellow men. Mr. Westinghouse was a splendid specimen of American manhood, slightly over six feet in height, and of fine presence, so that he was a striking and attractive figure anywhere. He was naturally amiable and a born diplomat, so that none who came in

contact with him ever failed to be agreeably impressed. He was, moreover, absolutely unaffected and straightforward in his intercourse with those about him, and easily accessible to anyone who wanted to see him. His memory was as marvelous as his other characteristics, and he never seemed to forget a name or a face where he has been in touch with the people long enough for an impression to be made.

He was very simple in his tastes and entirely devoid of ostentation. He always for long journeys traveled in his private car, which was really a traveling office, because his time was divided about equally between his office in New York and his home in Pittsburgh; but on local trips about Pittsburgh he used the ordinary day coach. While very far indeed from being what is popularly known as a "society man," he entertained a great deal when at his homes in Pittsburgh or Lenox and during the years when his home in Washington was open. During his long stays in Pittsburgh Mrs. Westinghouse was usually with him, and then their home was a scene of constant hospitality. He was so famous, and had such a wide acquaintance among eminent people of all countries that there was a constant succession of eminent visitors from our own country and abroad at his Pittsburgh home, and the guest books of the more than forty years of this period would be exceedingly interesting.

In this Journal of Naval Engineers it is fitting and interesting to note that Mr. Westinghouse was for a short time during the Civil War a volunteer Engineer Officer, entering in 1864 and resigning after the close of the war, in 1865. He was then under twenty years of age, but the inventive faculty was in evidence, and a marine governor, which was undoubtedly suggested by his experience while in the Navy, was one of his earliest inventions.

It is often a matter of wonder whether great men feel specially complimented by the honors which are conferred upon them by universities, learned societies and royalties after they have become so famous that nobody could possibly doubt it. However that may be, Mr. Westinghouse was the recipient

of a great many such honors. He had received three of the great engineering medals which are meant to represent that the recipient is considered one of the great men in engineering. These are the medals called after John Fritz, Edison and Grashoff. Numerous honorary degrees were conferred upon him, including that of Doctor of Engineering from the Royal Technical School at Charlottenburg and Doctor of Philosophy from Union College. He had also been decorated with the Orders of the Legion of Honor of France and of the Crown of Italy.

The esteem and affection in which Mr. Westinghouse was held by the officers and employees of his various companies is a striking tribute to his wonderful personality. It is often said that no man can be a hero to those who see him every day, but constant association with Mr. Westinghouse only served to emphasize his rare character. Something of his nature seemed to pass into his associates, and they acquired to some extent the virtues which he displayed in such a remarkable degree. Even the humble workers who hardly came in touch with him at all were nevertheless proud to work for so remarkable a man. His fame seemed in a way to affect and inspire every one of his men.

The writer has had the privilege of working for two great men, and of close association with them—Admiral George W. Melville, the famous Engineer-in-Chief of the Navy, and George Westinghouse, the great genius. He has always felt that their friendship was something which could not be too highly prized. It gave him a chance to see the workings of great natures at close range and to benefit by the inspiration which emanates from these specially gifted ones of Nature's children. He has found in talking with others who had been associated with these famous men that their feeling is the same, and that they count their years of service as among the golden days of their lives.

And just as it is a precious memory for those who were with him to rejoice in the personal association, so it is a pleasure for his contemporaries to feel that they were of his time and witnessed the birth and growth of the marvels of his

genius. All unite in a tribute of praise and admiration for the engineering genius and great benefactor of the race. In this panegyric, the writer would add his meed of affection and admiration for the great man who was his friend. He has endeavored to sketch the outlines of this great, heroic character for the benefit of those who did not know him personally, and as voicing the sentiments of the great industrial army which he commanded. He needs no monument, sculptural or literary, to perpetuate his memory, because wherever there is civilization and evidence of engineering, there, in his own works, the children of his brain, will be found monuments innumerable to George Westinghouse.

## EDWIN M. CORYELL, ASSOCIATE.

Edwin M. Coryell was born in Brooklyn, N. Y., September 25, 1847, and was educated in the public schools there. After three years as apprentice and draftsman, in 1869 he started out for himself as mechanical engineer, patent attorney and draftsman in New Haven, Conn. Two years later he secured employment at the Bessemer Steel Works, Troy, N. Y., where he came under the personal direction of Alexander L. Holley and was fellow draftsman with Allen Sterling. He served as draftsman at the Providence Steam Engine Works as well as on the Brooklyn Bridge, and from 1878 to 1879 was draftsman with the New York Gas Light Company. For the last 34 years Mr. Coryell had been connected with the A. S. Cameron Steam Pump Works, first as draftsman, then as salesman, assistant superintendent, superintendent and, finally, as consulting engineer. During that time he invented more than 200 patented devices for pumps and other machinery. He died suddenly on March 23, 1914.

## CAPTAIN RICHARD HENDERSON.

Captain Richard Henderson, U. S. Navy, Retired, member of the Society, died at his home, Salisbury, N. C., February 24, 1914.

## BOOKS RECEIVED.

LAYING OFF, OR THE GEOMETRY OF SHIPBUILDING, by EDWARD L. ATTWOOD and I. C. G. COOPER. LONGMANS, GREEN AND CO.; 119 pages; price, $1.75; 121 illustrations.

This work deals with the practical methods employed in the mold loft for laying off parts of ships' structure, and treats of the geometrical principles involved, and of their practical application. The principles of Descriptive Geometry involved are treated of in the first chapter; the following chapters give descriptions of the practical means employed.

The work is clearly written and well illustrated. It should be of great value to students in Naval Architecture, ships' draftsmen, as well as mold-loft operators.—H. C. D.

# NOTES OF ENGINEERING SOCIETIES.

## TECHNICAL LEAGUE OF AMERICA.

Suggestion has been made to form another society from the membership of large existing Engineering Societies for the purpose of discussing the material interest of the profession.

Suggestion has been made that the smaller Engineering Societies extend their activities in the direction of securing greater social recognition for the members of the profession.

*Extract from Manifesto.*

Engineers must realize that no one is going to raise their material status except themselves; we are not asking that one or two positions of phenomenal material advantage be created, upon which all engineers could set their gaze and tear each others' throats to reach. We are agitating that we may raise the material status of the profession so that all grades, engineer, designer, superintendent, checker, surveyor, draftsman, etc., may reap the benefit therefrom. This is a group matter and can only be achieved by group action.

Such action and organization as is here advocated would in no way interfere or clash with the activities of societies whose aims are the advancement of science and engineering. On the contrary, it is felt that the unselfishness and loyal devotion of engineers to their profession and the public interest is in no way better exemplified than by the suggestion made before one of the large national societies, to form another separate body rather than divert these bodies from their original purposes. The League would at all times be glad to see its members, who have reached positions of economic security, join the national society of the branch of engineering they belong to.

Let our aim be to place engineers in the foreground of pub-

lic opinion, place them in city, State and federal offices, commissions and boards, wherever they can best serve the public and by so doing increase the scope and prestige of the profession.

Is it not reasonable to say that if we place engineers in our legislative bodies, instead of having a government of legal technicalities we would secure constructive policies, with the expansion of public works, which would not only benefit the public, but would at the same time add opportunities to the whole profession in an ever-widening sphere?

Is it not reasonable to believe that with engineers in positions of social responsibility and control, and the profession united by a powerful economic organization, the material status of all grades from tracer upward would receive greater social consideration as well as greater material benefit?

The League cordially invites those who recognize the advent of the new era in engineering and who feel sympathy with any of the views herein expressed to send in their applications at once and take their stand on the platform offered in this manifesto.

The man who believes in a principle and supports it for that reason is the man of the highest mental and moral caliber, no matter what grade in the profession he has reached, and it is to such men that this address is particularly directed.

TECHNICAL LEAGUE OF AMERICA,
74 Cortlandt Street, New York, N. Y.

AN EFFICIENT CHECK VALVE.

# ASSOCIATION NOTES.

## SOCIETY BANQUET.

The banquet of the Society was held at the Army and Navy Club, Washington, D. C., April 25th, 1914. There were present 168 members and guests. The Society was very fortunate in having many distinguished guests from Congress, from the engineers in the commercial world, and from the naval service.

The Secretary of the Navy was the special guest of this occasion, and, though called to the White House during the course of the banquet, he was able to return in time to deliver his address.

The moving spirit of this banquet was an effort to secure better coöperation and unity of purpose between the naval service and the engineering personnel engaged in the production of naval material.

Lieutenant Commander Louis C. Richardson, U. S. N., acted as toastmaster.

In his introductory remarks he referred to the aims and purposes of the Society. The following is an extract from his remarks:

> I am pleased to see representatives here from so many fields of endeavor. You have come here to renew old acquaintanceships and abandon yourselves to the charm of an evening's amusement, but I feel that other motives and sentiments of a higher order have been the compelling influences that have brought you here.
>
> The society was founded for the purpose of developing naval engineering. An endeavor has been made to assemble at this banquet the men on whose united effort the progress of naval engineering depends. As little is known about the naval engineer, I will talk to you a few moments about him.
>
> The naval engineer, in order to be of the greatest usefulness to the country and a leader in the field of engineering, must be known for what he is to the administration; he must be in touch with the designing engi-

neers, inventors, and the manufacturers of engineering material in civil life, and his importance to the naval service must be thoroughly understood by all the officers of the Navy.

We have with us tonight distinguished representatives of the political world, leaders from the field of engineering in civil life, naval engineers—and I include with the naval engineers the engineers from that splendid corps, the Revenue Cutter Service, who have always had their machinery ready for the most hazardous undertakings, whether in peace or war.

Each of the representative bodies present has the same ultimate ideas. We are all working towards the same end—the giving to the people of America ships superior, from an engineering and material point of view, to any other ships in the world. With this in view, whether we are political, naval, or commercial, our professional and national pride, as well as our patriotism, makes it mandatory that we give to the nation the best ships at a reasonable cost.

The political must give us adequate funds with which to purchase the best material; the commercial must develop to a high state of efficiency the methods of production, so that the best and most modern material can be supplied at a reasonable cost; the naval engineer must construct, improve, and operate the engines so as to meet the best requirements of strategy and tactics.

The naval engineer feels the need of your coöperation. It is hoped that each annual banquet will see a greater representation from each branch and that we will, through a frank understanding of one another, be able to coördinate our efforts in such manner as to make each of us proud of the part we play in the development of naval engineering.

The naval engineer is not known or understood either in political or commercial life. In fact, his full importance is just beginning to be recognized by the Navy itself. He is a comparatively new factor and coincident with the development of the modern fleet. It has taken fifteen years for the Navy to understand the full importance of the amalgamation of the line with the engineers. Amalgamation was necessary in order to produce suitable officers to fill the positions of commanders of our fleets, squadrons, divisions and ships and to produce suitable officers to command our naval stations and to be the general managers of our manufacturing plants.

The naval engineer belongs to no separate branch or corps of the Navy. His aims are beyond and above making the engineering branch the most important one. He looks at the development of naval engineering in its true light, its relation to the hull, to strategy and tactics of war, and the allowances that must be made to permit the proportional development of the other elements of the ship. His training is broad and particularly fits him for all the important positions in our Navy.

The opening address was delivered by the President of the Society, Rear Admiral John R. Edwards, U. S. N. He referred to the wide scope that engineering plays in the con-

struction and operation of naval vessels and the widespread influence that engineering bears in the interrelation, being the Navy and the industrial activities of the country.

The toasts were as follows:

"The Engineer, the Modern Aladdin."
THE SECRETARY OF THE NAVY.
"The Shipbuilding Industry and Its Relations to Naval Power."
MR. HOMER L. FERGUSON, Vice President Newport News S. B. and D. D. Co.
"The Engineering Press as a Factor in Naval Development."
MR. C. W. BAKER, Editor *Engineering News.*
"Advanced Engineering Education Essential to Efficiency."
PROF. CHAS. LUCKE, Columbia University.
"Our Naval Program and the Good of the Navy."
HON. LEMUEL P. PADGETT, M. C.

Those attending had the pleasure of hearing Mr. Ed. J. Dooner, of Philadelphia, sing several Irish ballads, and Mr. William Murray rendered several of his most effective popular songs.

Short impromptu speeches were made by
The Hon. John J. Fitzgerald, M. C.
Hon. E. R. Mann, M. C.
Senator N. P. Byran.
Major General Geo. Barnett, U. S. M. C.

LETTER FROM CHIEF ENGINEER B. F. ISHERWOOD, U. S. N.

While the following letter was specifically addressed to the Members of the American Society of Naval Engineers to be read at their annual banquet, held April 25th at the Army and Navy Club, Washington, D. C., those who know of the work accomplished by ex-Engineer-in-Chief Isherwood, U. S. N., will fully understand that this communication was rather intended for that large and efficient complement of zealous officers who are now performing engineering duty ashore and afloat, and even for that much larger number of officers in the Navy who realize the importance of engineering efficiency and prestige.

Mr. Isherwood is now about 91 years of age, and until two years ago actively kept up his professional work. When measured by his scholarly attainments, extent and character of the engineering research work conducted by him, and his fame as a scientist, Mr. Isherwood has had no peer amongst the thousands who have performed engineering duty in the United States Navy as Commissioned Officers.

It is considered that the message will be an inspiration to those who know of the great worth of the man, for it comes from one who entered the service in 1844, and one who is the only living engineer officer who has personally observed every feature connected with the development of the steam Navy of the United States. When it is also remembered that for at least three decades this distinguished man was one of those foremost in promoting engineering advance, and that for two following decades he gave to the engineering world technical literature and monographs that are professional classics, the far-reaching influence of the message can be appreciated in its fulness.

111 East Thirty-Sixth St.,
New York City,
*April* 14th, 1914.

To the American Society of Naval Engineers.

Gentlemen:

It is a very old friend and Brother Engineer who greets you, one who is much enfeebled through sickness and the weight of years, but the request through Rear Admiral Edwards to send you all a message was couched in such words that I was much touched, and I make this effort to show you my appreciation.

There is one preëminent one, who has so lately passed, George Westinghouse, but I must leave to those enfeebled through illness and the advance of years the honor of pronouncing the panegyric on this splendid spirit.

His achievements have more eloquence than aught from tongue or pen.

My greeting to you is indeed from one of the Old Navy. I found the Engineer Corps, but as a weak infant; I gave of my best to make that infant grow into splendid manhood. It has been granted me to see those efforts crowned with an imperishable wreath, that wreath you, the Engineer Corps of the United States Navy.

All is passed now for me of effort or reward, it is for you to carry on, to push forward; and if my example can prove aught of inspiration to you, my young brothers of the New Navy, and to those still younger

brother engineers of the Future Navy, then shall I have done my little part.

There are many arts, all fine and splendid, but none has so benefited the world as the great Art of Engineering.

Therefore you Engineers be proud of your profession as I am proud to subscribe myself

Your Brother Engineer,

B. F. ISHERWOOD.

The address of Prof. Chas. Lucke is printed elsewhere in this issue.

The address of Mr. Homer L. Ferguson attracted considerable attention and has been treated of in editorials in the engineering press.

The following were present:

H. L. Aldrich, Ernest H. B. Anderson, Commander Martin Anderson, U. S. Navy.

Rear Admiral G. W. Baird, U. S. Navy; Lieutenant Colonal C. B. Baker, U. S. Army; C. W. Baker, H. A. Banks, Major General George Barnett, U. S. Marine Corps; Dr. Edmund Barry, Representative E. R. Bathrick, Captain E. P. Bertholf, U. S. Army, C. S.; Commander J. R. Y. Blakely, U. S. Navy; James B. Bonner, C. R. Bradbury, T. W. Brahany, Representative Fred A. Britten, Captain B. C. Bryan, U. S. Navy; Senator N. P. Bryan, Representative Thomas Butler.

William H. Callahan, Passed Assistant Engineer R. E. Carney, U. S. Navy, retired; R. H. Chappell, Lieutenant Alex. M. Charlton, U. S. Navy; Senator W. E. Chilton, Lieutenant C. R. Clark, U. S. Navy; M. S. Cooley, Rear Admiral A. R. Couden, U. S. Navy; Paymaster General T. J. Cowie, U. S. Navy; Lieutenant O. L. Cox, U. S. Navy; W. Griscom Coxe, Lieutenant Commander W. P. Cronan, U. S. Navy.

Secretary of the Navy Josephus Daniels, William L. De Baufre, Lieutenant Harvey Delano, U. S. Navy; Lieutenant Commander H. C. Dinger, U. S. Navy; Edward J. Dooner, E. H. Duff.

Rear Admiral John R. Edwards, U. S. Navy; E. S. Conti, Frank Emery, Colonel J. Ewing, Thomas Eynon.

Captain A. F. Fechteler, U. S. Navy; F. E. Ferguson, H. L. Ferguson, Rear Admiral B. A. Fiske, U. S. Navy; Representative John J. Fitzgerald.

S. Franklin Gardner, Fred J. Gauntlett, W. B. Gifford, H. A. Gillis, Lieutenant Commander J. B. Gilmer, U. S. Navy; C. P. Gleim, Colonel Glenn, U. S. Army; Engineer in Chief R. S. Griffin, U. S. Navy.

Captain David A. Hall, U. S. Revenue-Cutter Service; B. H. Hamilton, F. D. Herbert, H. T. Herr, H. L. Hibbard, H. DeBree Higgins, James S. Hill, Captain Frank T. Hines, U. S. Army; Lieutenant C. N. Hinkamp, U. S. Navy; Francis Hodgkinson, Commander U. T. Holmes, U. S. Navy; D. N. Hoover, O. Z. Howard, Major General C. F. Humphrey, U. S. Army, retired; M. R. Hutchison.

Lieutenant R. L. Irvine, U. S. Navy.

E. N. Janson, William N. Jeffress, J. E. Jenks, E. B. Johns, Archibald Johnson, Lieutenant Commander N. L. Jones, U. S. Navy.

Captain Gustav Kaemmerling, U. S. Navy; M. L. Katzenstein, W. D. Kearfott, Medical Inspector R. M. Kennedy, U. S. Navy; Passed Assistant Engineer William R. King, U. S. Navy, retired.

W. H. A. Lange, S. Inglis Leslie, George W. Littlehales, Leo Loeb, J. O. Leech, F. R. Low, Professor C. E. Lucke.

Engineer in Chief C. A. McAllister, U. S. Revenue-Cutter Service; Walter M. McFarland, John S. McKinney, H. A. Magoun, Representative James R. Mann, A. M. P. Maschmeyer, Jerry Mathews, Robert Mayo, Jr.; A. Le P. Mesney, J. F. Metten, James P. Mewshaw, Brigadier General Anson Mills, U. S. Army, retired; H. T. Morningstar, J. H. Mull, William Murray.

Assistant Secretary of the Treasury Byron R. Newton, Captain H. P. Norton, U. S. Navy.

J. C. O'Laughlin.

Representative L. P. Padgett, Civil Engineer C. W. Parks, U. S. Navy; Rear Admiral W. M. Parks, U. S. Navy; John Platt, Senator Miles Poindexter, J. W. Powell.

Director Joseph E. Ralph, Professor Walter Rautenstrauch, Alfred H. Raynal, Lieutenant Commander Louis C. Richardson, Representative Ernest W. Roberts, R. R. Row.

C. Melvin Sharpe, Senator William A. Smith, W. W. Smith, E. H. Sniffin, H. M. Southgate, Naval Constructor A. W. Stahl, U. S. Navy; Rear Admiral Homer Stanford, U. S. Navy; R. Paul Stout, Rear Admiral Joseph Strauss, U. S. Navy; Hal H. Smith.

Captain Emil Theiss, U. S. Navy; Senator J. R. Thornton, Representative J. Tribble.

General George Uhler.

Captain D. Vassilieff, I. R. N.; Leroy Vernon.

Ernest Walker, J. B. Walker, George E. Walton, Charles Ward, Chief Constructor R. M. Watt, U. S. Navy; Lieutenant D. A. Weaver, U. S. Navy; Commander W. W. White, U. S. Navy; Captain H. A. Wiley, U. S. Navy; Representative W. E. Williams, Lieutenant E. E. Wilson, U. S. Navy; Lieutenant C. C. Windsor, U. S. Navy; Lieutenant Commander H. T. Winston, U. S. Navy.

W. E. Yelverton.

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THE FOLLOWING MEMBERS AND ASSOCIATES have joined the Society since the publication of the February, 1914, number of the Journal.

MEMBERS.

Atkins, Alfred W., Lieutenant, U. S. N.

Bartlett, Owen, Lieutenant, U. S. N.

Bastedo, Paul H., Lieutenant, U. S. N.

Bidwell, Abel T., Lieutenant, U. S. N.
Boyden, Donald, Ensign, U. S. N.
Bryan, George S., Lieutenant, U. S. N.
Cochran, Walker, Ensign, U. S. N.
Day, George C., Commander, U. S. N.
Doherty, Stephen, Lieutenant, U. S. N.
Donald, Harry G., Lieutenant, U. S. N.
Duncan, Paul B., Lieutenant Commander, U. S. N.
Fulton, Garland, Ensign, U. S. N.
Griswold, Ralph M., Lieutenant, U. S. N.
Gulbranson, Clarence, Ensign, U. S. N.
Haggart, Robert S., Ensign, U. S. N.
Hill, Kinchen L., Ensign, U. S. N.
Jeans, Howard S., Ensign, U. S. N.
Jessop, Earl P., Lieutenant Commander, U. S. N.
Kilpatrick, Walter K., Lieutenant, U. S. N.
Kurtz, Thomas R., Lieutenant Commander, U. S. N.
Larimer, Marc W., Ensign, U. S. N.
Lofquist, Emanuel A., Lieutenant, U. S. N.
McCormack, Harvey W., Lieutenant, U. S. N.
Mecleary, Howard B., Lieutenant, U. S. N.
Newton, William F., Lieutenant, U. S. N.
Ogan, Joseph V., Lieutenant, U. S. N.
Pace, Ernest M., Ensign, U. S. N.
Patch, Ernest L., Ensign, U. S. N.
Rice, Arthur H., Lieutenant, U. S. N.
Saunders, Harold E., Ensign, U. S. N.
Spafford, Edward E., Lieutenant Commander, U. S. N.
Stevens, James G., Lieutenant, U. S. N.
Sypher, Jay H., Commander, U. S. N.
Weaver, W. D., Charlotteville, Va.
Weems, Philip V. H., Ensign, U. S. N.
Wilson, Eugene E., Lieutenant, U. S. N.

ASSOCIATES.

Ansell, Henry A., Vice President, Pennsylvania Flexible Metallic Tubing Co., Cor. Broad and Race Sts., Philadelphia, Pa.

Eckliff, James C., 326 and 328 Ford Building, Detroit, Michigan.

Frank, Dr. Karl, Rooms 1017–1020 West Street Building, 90 West Street, New York City.

Kellogg, James W., Manager, Dept. of Marine Sales, General Electric Company, Schenectady, N. Y.

Kemmer, James E., Machinist, U. S. N.

Mardaga, Louis, Naval Engineering Experiment Station, Annapolis, Md.

Williams, Eugene B., Manager, Marine Department, B. F. Sturtevant Company, Hyde Park, Mass.

Hoxsie, Lieut. Robert F., U. S. N.

Hull, Raymond L., Ensign, U. S. N.

Ingram, Howard E., Ensign, U. S. N.

Irwin, Noble E., Ensign, U. S. N.

Jones, Earl W., Lieutenant-Commander, U. S. N.

Kilpatrick, Walter G., Lieutenant, U. S. N.

Kurtz, Thomas K., Lieutenant-Commander, U. S. N.

Lattimer, Mark W., Ensign, U. S. N.

Leftwich, Rossmore A., Lieutenant, U. S. N.

McCormick, Farrow W., Lieutenant, U. S. N.

Meehan, Howard P., Lieutenant, U. S. N.

Newton, William G., Lieutenant, U. S. N.

Ogan, Joseph V., Lieutenant, U. S. N.

Pace, Ernest M., Ensign, U. S. N.

Patch, Edson L., Ensign, U. S. N.

Rice, Arthur N., Lieutenant, U. S. N.

Saunders, Harold E., Ensign, U. S. N.

Stafford, Edward S., Lieutenant-Commander, U. S. N.

Stevens, James C., Lieutenant, U. S. N.

Sypher, Jay H., Commander, U. S. N.

Weaver, W. S., Charlottesville, Va.

Irvine, Philip V. H., Ensign, U. S. N.

Wilson, Eugene E., Lieutenant, U. S. N.

Alexandria.

Ansell, Henry A., Vice-President, Pedrazzini Flexible Metallic Tubing Co., Cor. Broad and Race Sts., Philadelphia, Pa.

Baltimore, Md.

Frank, Dr. Karl, Rooms 1017-1020 West Street Building, 90 West Street, New York City.

Keller, James W., Manager, Dept. of Marine Sales, General Electric Company, Schenectady, N. Y.

Kenner, James E., Machinist, U. S. N.

Mardaga, Louis, Naval Engineering Experiment Station, Annapolis, Md.

Williams, Eugene B., Manager, Marine Department, R. F. Stuttevant Company, Hyde Park, Mass.

Hopkins, James C., 326 and 328 Ford Building, Detroit, Michigan.

## ERRATA.

MAY, 1914, NUMBER OF JOURNAL.

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"Advanced Engineering Education," page 509. The name of Prof. Chas. Lucke appeared as Lüeke. This is an error very much regretted.

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Description and Trials of Torpedo-Boat Destroyer *Duncan*, page 548. Table I, line 8, Columns 2, 3, 4, Total Nozzle areas should read 19.85 — 27.79 — 39.18.